

AD-A153 779

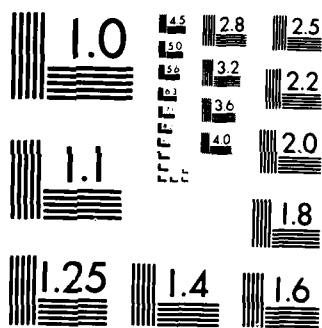
COMPUTER-AIDED THERMOHYDRAULIC DESIGN OF TEMA TYPE E
SHELL AND TUBE HEAT. (U) ARMY MILITARY PERSONNEL CENTER
ALEXANDRIA VA N J KOLAR 01 APR 85

1/2

UNCLASSIFIED

F/G 13/1

NL



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

AD-A153 779

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

| REPORT DOCUMENTATION PAGE | | READ INSTRUCTIONS BEFORE COMPLETING FORM |
|--|--|---|
| 1. REPORT NUMBER | 2. GOVT ACCESSION NO. | 3. RECIPIENT'S CATALOG NUMBER |
| | AD-A153779 | |
| 4. TITLE (and Subtitle) Computer-aided thermohydraulic design of TEMA type E shell and tube heat exchangers for use in low pressure, liquid to liquid, single phase applications. | 5. TYPE OF REPORT & PERIOD COVERED Final Report 1 April 1985 | 6. PERFORMING ORG. REPORT NUMBER |
| 7. AUTHOR(s) CPT Nicholas J. Kolar | 8. CONTRACT OR GRANT NUMBER(s) | |
| 9. PERFORMING ORGANIZATION NAME AND ADDRESS Student HQDA, MILPERCEN (DAPC-OPA-E) 200 Stovall Street Alexandria, Va. 22332 | 10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS | |
| 11. CONTROLLING OFFICE NAME AND ADDRESS HQDA, MILPERCEN, ATTN: DAPC-OPA-E 200 Stovall Street Alexandria, Va. 22332 | 12. REPORT DATE 1 April 1985 | 13. NUMBER OF PAGES 168 |
| 14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office) | 15. SECURITY CLASS. (of this report) | 15a. DECLASSIFICATION/DOWNGRADING SCHEDULE |
| 16. DISTRIBUTION STATEMENT (of this Report) Approved for public release; distribution unlimited. | | |
| 17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report) | | |
| 18. SUPPLEMENTARY NOTES Document is a thesis from Bucknell University, Lewisburg, Pa. 17837. | | |
| 19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Shell and tube heat exchanger design Computer-aided design | | |
| 20. ABSTRACT (Continue on reverse side if necessary and identify by block number) Classification, nomenclature, utilization and cost estimating of shell and tube heat exchangers are presented along with an historical overview of various methods currently employed in their design. A procedure for providing estimates of shell and tube heat exchanger design is developed in detail along with a computer program which employs this sizing algorithm for low pressure liquid-to-liquid exchanger applications. Additionally, problems encountered in the design and manufacture of shell and | | |

DD FORM 1 JAN 73 1473

EDITION OF 1 NOV 65 IS OBSOLETE

SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

85 64 15 112

continued from block 20;

tube heat exchangers are described along with present methods of solution for each.

2015
COPY
2-2-2015
3

[illegible]

A1

A1

A thesis submitted to the faculty of Bucknell University in
partial fulfillment of the requirements for the degree of
Masters of Science in Mechanical Engineering.

I, Nicholas J. Kolar, do grant permission
for my thesis to be copied.

Nicholas John Kolar

COMPUTER-AIDED THERMOHYDRAULIC DESIGN OF TEMA TYPE E
SHELL AND TUBE HEAT EXCHANGERS FOR USE IN LOW PRESSURE,
LIQUID-TO-LIQUID, SINGLE PHASE APPLICATIONS.

BY

NICHOLAS JOHN KOLAR

PRESENTED TO THE FACULTY OF
BUCKNELL UNIVERSITY
IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF
MASTERS OF SCIENCE IN MECHANICAL ENGINEERING

APPROVED:

J. Ben Austin
Advisor

James W. Zaiser
Department Chairperson

1 April 1985

ACKNOWLEDGEMENTS

The writer is especially indebted to the following individuals:

My advisor, Professor J. Ben Austin, Jr., for his instruction, suggestions and advice.

Professors Charles H. Coder, James N. Zaiser, and Thomas P. Rich for their encouragement and advice.

Mr. A. Tim Chase, Manager Special Products Division of Struthers Wells Corporation and Chairman of TEMA Technical Committee, for his professional critique, in-depth discussion of current shell and tube heat exchanger design, and detailed tour of the Struthers Wells Manufacturing facilities.

Ms. Christa Decker for her cooperation and expert assistance in the use of the CAED Laboratory.

Ms. Kimberly Lazar for her cheerful and exceptionally efficient typing of this thesis.

Lastly, but by no means the least, my family, and in particular my wife Barbara, whose understanding, consolation and encouragement has been both loving and unswerving.

TABLE OF CONTENTS

| | <u>Page</u> |
|--|-------------|
| I. Introduction | 1 |
| II. Basic Components of Shell and Tube Heat Exchangers | 3 |
| A. General | 3 |
| B. Tubes | 3 |
| C. Tubesheets | 3 |
| D. Shell | 5 |
| E. Shell Nozzle | 5 |
| F. Tube-side Channels and Nozzles | 5 |
| G. End Covers | 5 |
| H. Pass Partition Plate | 6 |
| I. Baffles | 6 |
| III. Codes and Standards | 9 |
| A. General | 9 |
| B. ASME Boiler and Pressure Vessel Code | 9 |
| C. Foreign Pressure Vessel Codes | 9 |
| D. ANSI Standards | 10 |
| E. API Standards | 10 |
| F. HEI Standards | 10 |
| G. TEMA Standards | 10 |
| IV. Design Techniques | 13 |
| A. General | 13 |
| B. Delaware Method | 15 |
| C. Bell Method | 15 |

TABLE OF CONTENTS (CONT'D)

| | <u>Page</u> |
|---|-------------|
| D. Tinker Method | 16 |
| E. Devore Method | 19 |
| F. Numerical Methods | 19 |
| G. Proprietary Programs. | 20 |
| V. Shell and Tube Exchanger Utilization. | 23 |
| VI. Problems Encountered in Shell and Tube Heat Exchanger . | 24 |
| A. General | 24 |
| B. Flow Induced Vibrations | 24 |
| C. Fouling | 25 |
| VII. Computer-Aided Design Procedure | 28 |
| A. General | 28 |
| B. Instructions. | 28 |
| C. Input Data. | 29 |
| D. Number of Tubes Calculations. | 31 |
| E. Available Tube Surface Area | 35 |
| F. Maximum and Minimum Tube-Side Flow Rates. | 35 |
| G. Maximum and Minimum Shell-Side Flow Rates | 37 |
| H. Thermo-Hydraulic Properties of the Tube Fluid . . . | 39 |
| I. Shell-Fluid Exit Temperature Determination. | 40 |
| J. Calculation of Maximum and Minimum Capacity Rates . | 41 |
| K. Heat Transfer Effectiveness Calculation | 42 |
| L. Determination of Number of Transfer Units | 42 |

TABLE OF CONTENTS (CONT'D)

| | <u>Page</u> |
|--|-------------|
| M. Calculation of Estimated Overall U-Value. | 43 |
| N. Determination of Possible Exchanger | 45 |
| Configurations | |
| O. Output of Possible Exchanger Configurations | 56 |
| P. User Selection of Possible Exchanger. | 57 |
| Configurations | |
| Q. Calculation of Estimated Capital Cost | 57 |
| R. Final Design Output | 59 |
| S. User Directed Changes to Input Parameters | 60 |
| VIII. Example Problem | 61 |
| IX. Computer Results. | 62 |
| X. Conclusion. | 77 |
| A. Discussion. | 77 |
| B. Suggestions for Improvement | 78 |
| XI. References. | 79 |
| XII. Appendix A: Variable Listing | |
| Appendix B: Main Program Listing | |
| Appendix C: Fluid Property Subroutines (Water Only) | |
| Appendix D: PRESCK Subroutine | |
| Appendix E: TFRICT Subroutine | |
| Appendix F: SFRICT Subroutine | |
| Appendix G: Print Subroutines | |

LIST OF FIGURES

| <u>Figure</u> | | <u>Page</u> |
|---------------|---|-------------|
| 1 | Diagram of Typical Fixed Tube Sheet Heat Exchanger. . . | 4 |
| 2 | Partition Plate Configurations. | 7 |
| 3 | Baffle Configurations | 8 |
| 4 | TEMA Type Designations. | 12 |
| 5 | Leakage Paths for Flow Bypassing the Tube Matrix. . . . | 14 |
| 6 | Bell Method Design Chart. | 17 |
| 7 | Standard Tube Layout Designs. | 32 |
| 8 | Effectiveness - NTU Chart for a 1-2 Exchanger | 44 |

| <u>Table</u> | | <u>Page</u> |
|--------------|---|-------------|
| 1 | Typical Overall Design Coefficients for Shell and Tube Heat Exchangers | 46 |

ABSTRACT

Classification, nomenclature, utilization and cost estimating of shell and tube heat exchangers are presented along with an historical overview of various methods currently employed in their design.

A procedure for providing preliminary estimates of shell and tube heat exchanger design is developed in detail. The author formulates a computer program which employs this sizing algorithm for low pressure liquid-to-liquid heat exchanger applications.

Additionally, problems encountered in the design and manufacture of shell and tube heat exchangers are described along with present methods of solution for each.

I. INTRODUCTION

In the most basic terms, a heat exchanger is a device which provides for the transfer of thermal energy between fluids at different temperatures. Since high energy costs reduce profit margins of industrial products, any means of recovering by-product thermal energy, may result in an increase in corporate gains. For this reason, heat exchangers of all shapes, sizes, and configurations can be found throughout a wide range of industries.

Because of several factors which include: large ratios of heat transfer area to volume and weight, ease of construction and maintenance, design flexibility, and availability of proven design methodology, shell and tube heat exchangers are the most widespread and commonly used basic heat exchanger configuration in the process industries.[1]* It is with this in mind that shell and tube heat exchangers were chosen as the subject for this computer-aided design.

It should be noted that the design of a shell and tube exchanger consists of two basic areas. The first is the thermal-hydraulic design, which considers process fluids, flow arrangements, fluid flow rates, inlet and outlet fluid temperatures, fouling factors, pressure drops, construction materials and overall dimensions to ensure that the design allows for the thermal energy transfer required in the problem. The second is the mechanical design of the heat exchanger which ensures the mechanical integrity of the exchanger components during its operation. Pressure stresses and thermal stresses must be evaluated for each integral component, along with other considerations such as methods or

* Numbers in the [] bracket indicate sitings in the References.

B. Delaware Method

Probably the most widely publicized research program in the area of shell and tube heat exchanger design was an ASME sponsored project undertaken at the University of Delaware from 1947 to 1963. Under the auspices of Professors Olaf Bergelin and Allan Colburn, the Department of Chemical Engineering did extensive experimental measurements of shell-side heat transfer and pressure drop for different configurations of shell and tube heat exchangers. Systematically, baffle spacing, baffle-to-shell clearances, and tube-to-baffle clearances were adjusted to measure their effect on the shell-side performance characteristics. The Delaware Project produced several technical papers, and provided invaluable experimental data. One quantitative result of the project was an equation for the heat transfer coefficient which includes five correction factors to the ideal cross-flow heat transfer coefficient. Another, was an equation for the shell-side pressure drop which includes three correction factors to the ideal pressure loss of a shell with no leakage or bypass.

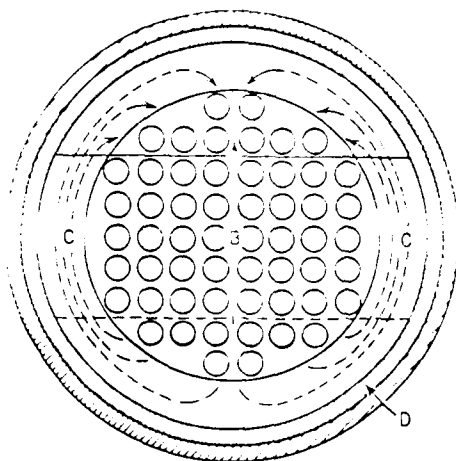
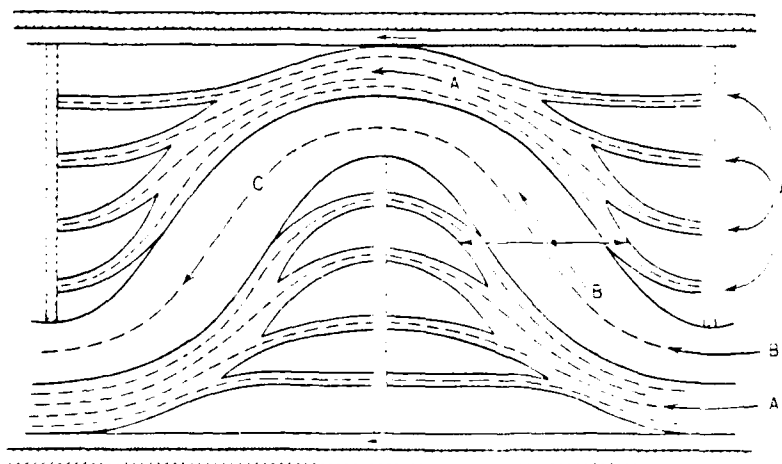
It must be noted, that although this method is generally accepted to be the best available in the open literature, it is not extremely accurate. Tests have shown that this method predicted shell-side heat transfer coefficients from about 50 percent low to 100 percent high, and the predicted shell-side pressure loss was from about 50 percent low to 200 percent high.[10]

C. Bell Method

Professor Kenneth J. Bell from Oklahoma State University devised a method for preliminary design of shell and tube heat exchangers which

FIGURE 5

LEAKAGE PATHS FOR FLOW BYPASSING THE TUBE MATRIX

FRONT VIEWSIDE VIEW

IV. DESIGN TECHNIQUES



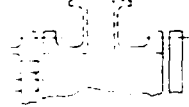
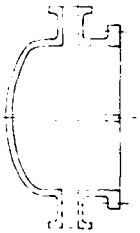

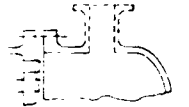
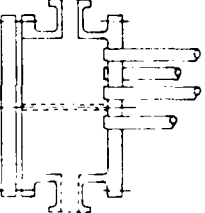

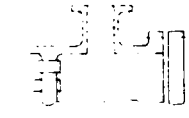
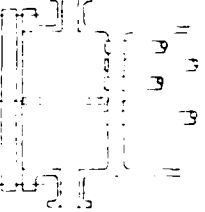


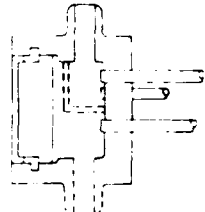

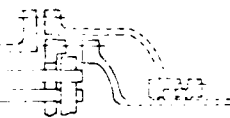
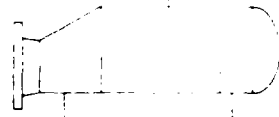
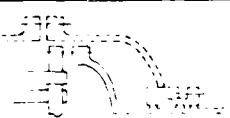

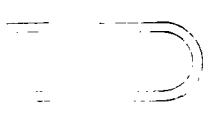
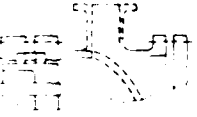
A. General

The most dynamic period of development in heat transfer occurred between 1920 and 1950. During this period almost all of the principles of thermodynamic behavior of heat exchangers became understood. One book, which was published after this important technological period, stands alone in the design of shell and tube heat exchangers. The book, Process Heat Transfer, by Donald Q. Kern, a professor from Case Institute of Technology, has provided the methods and procedures for the design of more shell and tube heat exchangers than any other single source.[8]

Probably the most difficult design parameters to quantify are the shell-side performance characteristics. Ideally, the shell-side flow can be viewed as a cross flow of fluid across a bank of tubes. But in reality, the flow goes both across the tubes and parallel to the tubes due to the serpentine motion of the fluid caused by the internal baffles of the exchanger. This is further complicated by leakage which occurs due to gaps between the baffle and the shell and spaces between the baffle and the tubes. This leakage or bypass flow causes a decrease in the heat transfer coefficient and uncertainty in the shell-side pressure drop. A diagram of the leakage paths taken from Reference 9 is shown in Figure 5. The elusive shell-side performance characteristics have been the cause for many independent studies, ultimately resulting in innovative methods for design.

FIGURE 4

TEMA TYPE DESIGNATIONS

| FRONT END STATIONARY HEAD TYPES | | SHELL TYPES | | REAR END HEAD TYPES | |
|------------------------------------|--|-------------|---|------------------------|--|
| A |  CHANNEL AND REMOVABLE COVER | E |  ONE PASS SHELL | L |  FIXED TUBESHEET LIKE "A" STATIONARY HEAD |
| B |  BONNET (INTEGRAL COVER) | F |  TWO PASS SHELL WITH LONGITUDINAL BAFFLE | M |  FIXED TUBESHEET LIKE "B" STATIONARY HEAD |
| C |  REMOVABLE TUBE BUNDLE ONLY CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER | G |  SPLIT FLOW | N |  FIXED TUBESHEET LIKE "N" STATIONARY HEAD |
| N |  CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER | H |  DOUBLE SPLIT FLOW | P |  OUTSIDE PACKED FLOATING HEAD |
| D |  SPECIAL HIGH PRESSURE CLOSURE | J |  DIVIDED FLOW | S |  FLOATING HEAD WITH BACKING DEVICE |
| | | K |  KETTLE TYPE REBOILER | T |  PULL THROUGH FLOATING HEAD |
| | | X |  CROSS FLOW | U |  U-TUBE BUNDLE |
| | | | | W |  EXTERNALLY SEALED FLOATING TUBESHEET |

and abroad. It has proven successful over the years because it is based on sound engineering practice. Probably most notable is the standardization of size designation and nomenclature which is used throughout the industry. Figure 4 shows the standard letter designations for the stationary head types, shell types, and rear-end head types. The program presented in this thesis designs TEMA "E" shell configurations.

TEMA standards require that ASME codes are followed. TEMA does cover internal components where ASME code rules are not available or are not applicable. In essence, TEMA provides good solid design and construction rules for heat exchangers considered as a special type of pressure vessel.

Three different classes of heat exchangers are developed in detail.

Class "R" - Shell and tube heat exchanges which must meet the severe requirements of the petroleum industry.

Class "C" - Shell and tube heat exchangers which must meet the moderate requirements of commercial and general process applications.

Class "B" - Shell and tube heat exchangers which must meet the requirements of chemical process service.

For each classification, TEMA sets the standards for design for each of the integral parts of the heat exchanger. These include: tubing, shells, baffles, heads, gaskets, tubesheets, channels, nozzles, end covers and bolting.

Additionally, TEMA provides information on generally used materials, fouling resistances and physical properties of fluids. General information such as characteristics of tubing, metal properties and conversion factors are also provided.

There is presently international interest in developing a code which will provide rules for use on an international basis. There are several major obstacles to its development including safety factors and material specifications.

D. American National Standards Institute (ANSI) Standards

ANSI is a Federal organization which manages standards throughout the United States. The ASME Code is an ANSI standard. A particular ANSI standard that is not covered by any other code is one that deals specifically with chemical processes.

E. American Petroleum Institute (API) Standards

API standards deal specifically with shell and tube heat exchangers which are used in connection with oil refinery services. The standards have been developed over the years from the accumulated knowledge and experiences of petroleum refiners nationwide.

F. Heat Exchange Institute (HEI) Standards

The HEI organization has developed standards for heat transfer equipment for use in steam power plants. Particular areas include: standards for closed feedwater heaters, standards for power plant heat exchangers (electric generating plants), and standards for steam surface condensers.

G. Tubular Exchanger Manufacturer Association (TEMA) Standards

This standard was established in 1939 in order to provide a measure of quality as well as uniformity in the design and construction of shell and tube heat exchangers. It is used extensively in the United States

III. CODES AND STANDARDS

A. General

Shell and tube heat exchangers are the most versatile and most widely used type of heat transfer equipment. Units can be designed to handle sensible heating, condensing and boiling. Operating pressures range from near vacuum to above 5800 psi. They may weigh as much as 200 metric tons and encompass as much as 50,000 ft² of heat transfer surface area.[7] To find codes and standards for its complex design, one needs to go to several different sources.

B. American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code

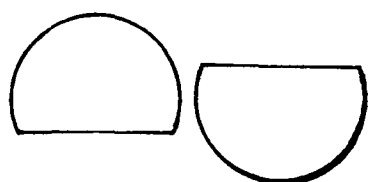
A shell and tube heat exchanger is indeed a pressure vessel and therefore must comply with several sections of this code which apply to: power boilers, material specifications, nuclear power plant components, nondestructive examination, pressure vessels, and welding and brazing qualifications.

In the United States and Canada, the ASME code is recognized as the means by which control over the design, fabrication and testing of boilers and pressure vessels is achieved.

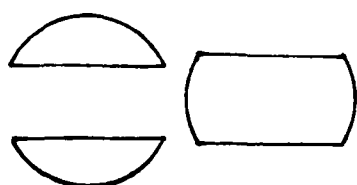
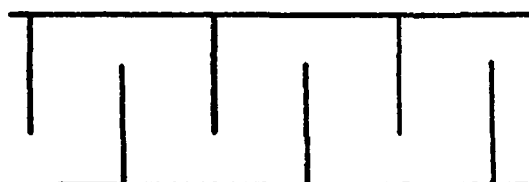
C. Foreign Pressure Vessel Codes

Most major industrial countries such as France, Great Britain, and West Germany have their own boiler and pressure vessel codes. Normally, if a unit is to be placed in service in one of these countries, their standards and limitations must be adhered to.

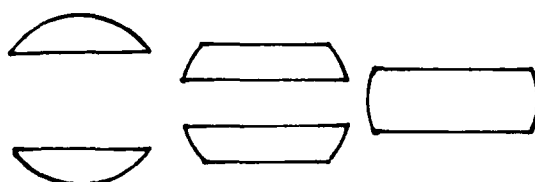
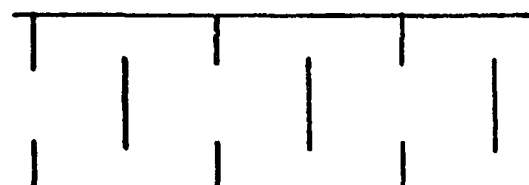
FIGURE 3
BAFFLE CONFIGURATIONS



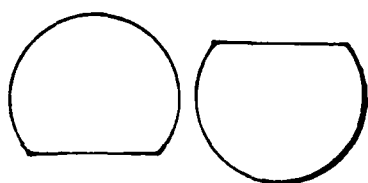
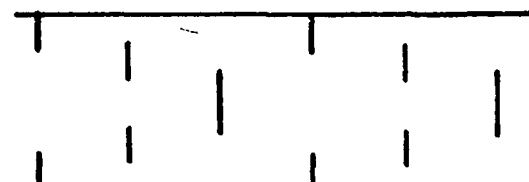
SINGLE SEGMENTAL



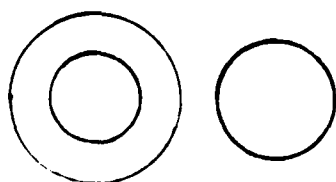
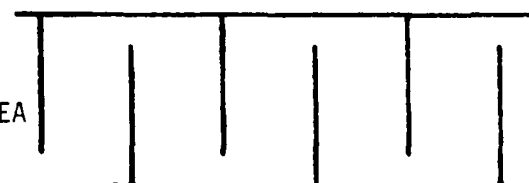
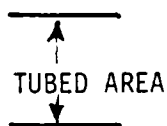
DOUBLE SEGMENTAL



TRIPLE SEGMENTAL



SEGMENTAL - NO TUBES IN WINDOW



DISK AND DONUT

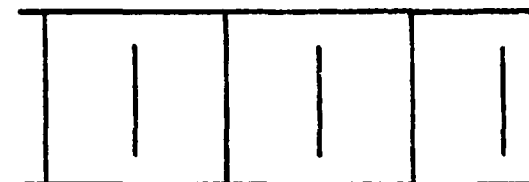
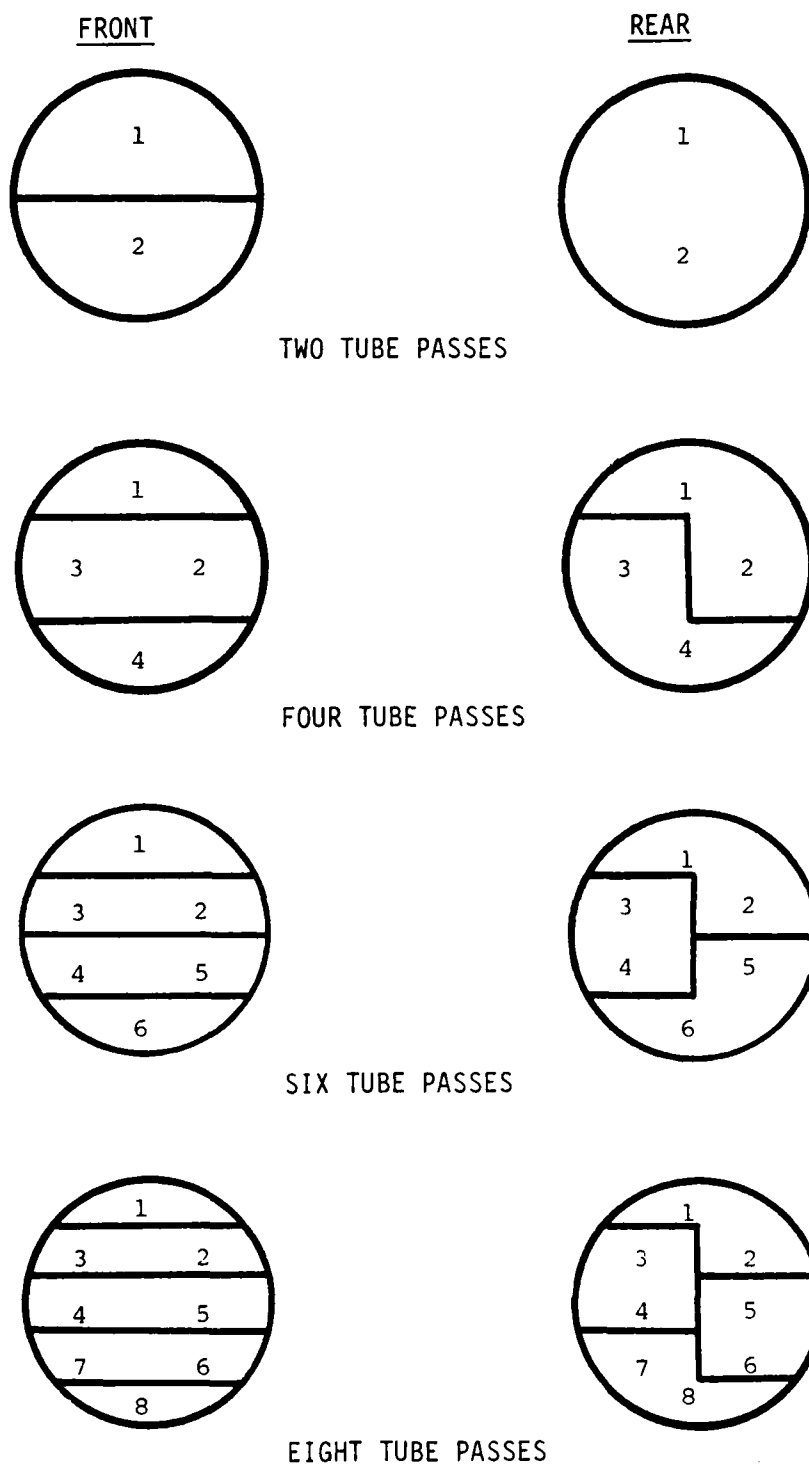


FIGURE 2
PARTITION PLATE CONFIGURATIONS



The most common configurations are: one shell pass two tube passes, one shell pass four tube passes, one shell pass six tube passes, and one shell pass eight tube passes. In these multiple-tube pass heat exchangers, pass partitions are required in the tube-side chambers of the heat exchanger to direct the flow of the tube fluid. The arrangement of the tube-side pass partitions affects the heat transfer behavior of the heat exchanger, and obviously affects the placement of the tube fluid entrance and exit nozzles. The most common tube-side pass partitions are shown in Figure 2.[3,4]

I. Baffles

The interior baffles of the heat exchanger serve two important functions. First, they support the tube bundle keeping the proper tube spacing during both construction and operation and at the same time prevent unwanted vibration in the tube bundle. Secondly, they provide a path for the shell-side fluid to move back and forth across the tube bundle increasing the shell-side fluid velocity and also increasing the heat transfer coefficient.

Several of the baffle configurations that are in use today are shown in Figure 3. The most common (and the one that this computer-aided design program is based upon) is the single segmental. The segment sheared off must be less than 50 percent and for liquid to liquid applications, a baffle cut of 20 - 25% is common.[6]

are: roller expansion of the tube into ungrooved tubesheets, hydraulic expansion of the tubes into grooved tubesheets, and welding the tube ends directly to the tubesheets.

D. Shell

The shell is a cylindrical vessel which holds the shell-side fluid. In small diameter shells (less than 24 inches), the shell is cut from standard pipe. For larger diameter shells, the shell is fabricated from rolling metal sheets of the desired dimensions and longitudinally welding the butt joint.

E. Shell Nozzle

The shell has two nozzles which allow for the entrance and exit of the shell-side fluid. The inlet nozzle normally also has an impingement plate which prevents the incoming fluid stream from impacting directly upon the exposed tubes causing unwanted erosion and vibration.

F. Tube-side Channels and Nozzles

The tube-side channels and nozzles direct and control the flow of the tube-side fluid when it is not in the tubes of the heat exchanger.

G. End Covers

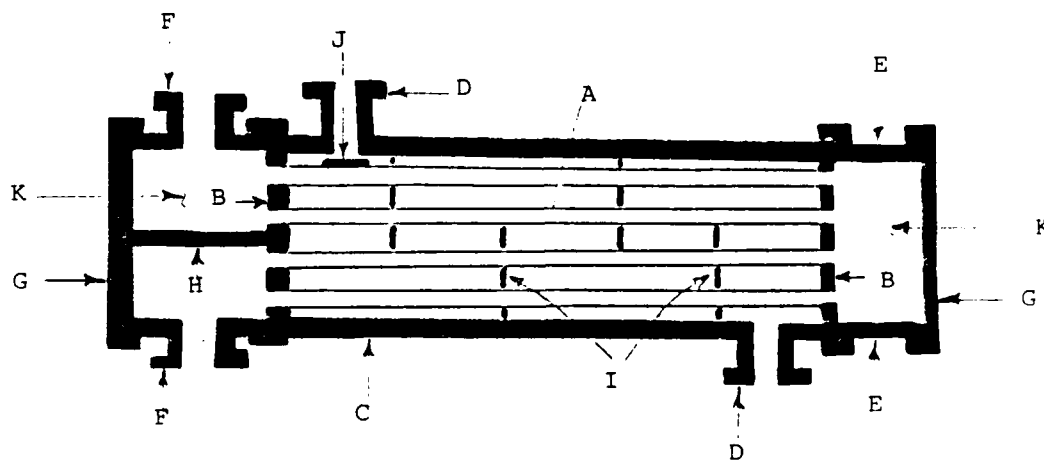
The end covers are metal plates that normally are bolted onto the ends of the shell. They are bolted rather than welded to facilitate maintenance and inspection of the tubes and channels in the heat exchanger.

H. Pass Partition Plate

The fluid in the tubes may travel the length of the exchanger more than one time. These are heat exchangers with multiple tube passes.

FIGURE 1

DIAGRAM OF A TYPICAL FIXED TUBE SHEET SHELL AND TUBE HEAT EXCHANGER



- A. TUBES
- B. TUBESHEETS
- C. SHELL
- D. SHELL-SIDE NOZZLE
- E. TUBE-SIDE CHANNEL
- F. TUBE-SIDE NOZZLE
- G. END COVER
- H. PASS PARTITION PLATE
- I. BAFFLE
- J. IMPINGEMENT PLATE
- K. TUBE-SIDE CHAMBER

II. BASIC COMPONENTS OF SHELL AND TUBE HEAT EXCHANGERS

A. General

For the purpose of establishing standard terminology, Figure 1 illustrates several basic components of shell and tube heat exchangers which will be described later. Even though there are thousands of specific design features that can be used, these specific components will invariably be present in all shell and tube heat exchangers.

B. Tubes

The tubes are the most basic component of the shell and tube heat exchanger. The tube wall provides the heat transfer surface between the fluid flowing inside the tube and the fluid flowing inside the shell. The tubes can be either seamless or welded. The metals used are carbon steel, low alloy steel, stainless steel, copper and copper alloy, nickel and nickel alloy, aluminum and aluminum alloy, titanium and zirconium, although special metals may be used to meet specific applications.[2]

The tubes may be bare or have integral circumferential finning. The computer-aided design program formulated in this thesis assumes that the tubes used are bare. The assemblage of all the tubes in the heat exchanger is referred to as the tube bundle.

C. Tubesheets

The tubes of the exchanger are held in place by being inserted into holes in metal sheets at either end of the shell. They are fixed in place by a variety of methods. The tube to tubesheet joint has the single objective of sealing against interleakage between the tube-side and shell-side fluids. Common methods of joining the two components

bonding components, types and sizes of gaskets and seals, and minimization of flow induced vibrations. This thesis covers only the thermal-hydraulic design of the heat exchanger. Although equally important as the thermal-hydraulic design, the mechanical design is much more intricate and has been left to those who are most capable of accomplishing the task - namely, the shell and tube heat exchanger manufacturers of the United States and abroad. For it is their decades of experience, experimentation, and research that has developed the standards of construction and manufacture that are in use today.

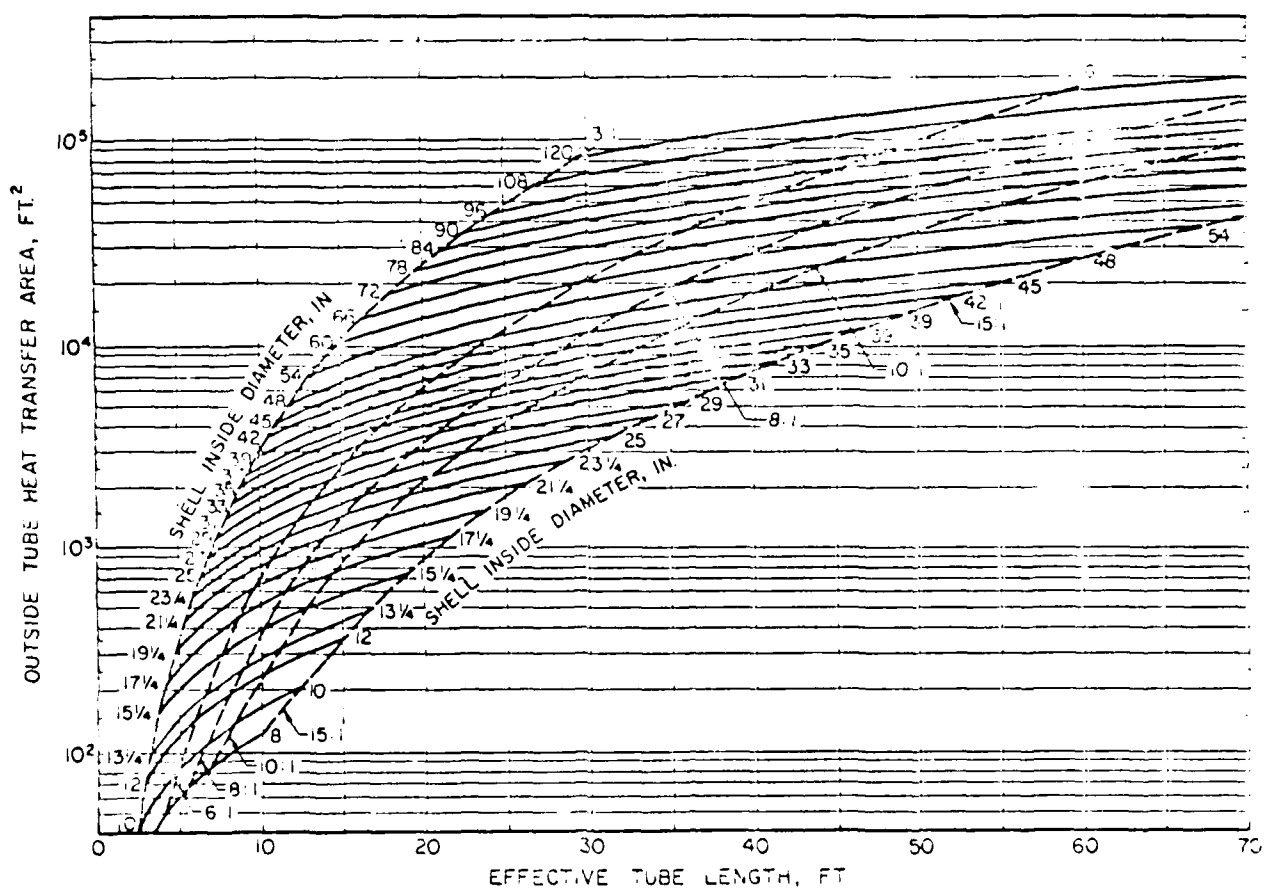
also takes into account deviations from ideality. His method is unlike the Delaware method in that he does not directly calculate the shell-side heat transfer coefficient. He uses typical overall design coefficients (U -values) for shell and tube heat exchangers depending upon the type of fluids that are used. The data has been collected from years of experimental performance and commercial usage. Using this overall design coefficient, Bell calculates the required outside tube heat transfer area. The key to the Bell Method is the graph shown in Figure 6. Bell has depicted on this graph the possible configurations of shell and tube exchangers using standard tube counts for $3/4$ inch O.D. tubes on a $15/16$ inch triangular pitch for a fixed tube sheet exchanger with one tube-side pass. Also shown are the effective tube length to shell diameter ratios (3:1, 5:1, etc.) available. The user simply enters the ordinate of the graph with the required area and immediately knows the combinations of tube length and shell diameter that will provide that area in a single shell for an exchanger of Bell's standard tube size and layout. For those exchangers that do not conform to Bell's standard, three correction factors to the area calculation are utilized; one for different tube diameters and layouts, one for different numbers of tube-side passes, and one for different TEMA tube bundle construction types. The Bell Method is used solely for preliminary estimates of the heat exchanger size, and is obviously not a complete design methodology.

D. Tinker Method

In 1951 Townsend Tinker, vice-president for engineering of Ross Heat Exchanger Division, American Radiator and Standard Sanitary Corporation

FIGURE 6

BELL METHOD DESIGN CHART



was the first to develop a complex but effective methodology for including the effects of deviations from ideality. The fluid streams in the shell depicted in Figure 5 was taken from his original work. Tinker uses graphs of modified shell-side Reynolds numbers vs. shell-side fluid friction factors and modified shell-side Reynolds numbers vs. shell-side heat transfer coefficients for three different tube-hole patterns: triangular pitch, square pitch and staggered square pitch. The experimental data he uses comes from the research done by the Delaware Project.

While Tinker's basic derivations are general, they are cumbersome unless the range of possible combinations of his geometric parameters is reduced. He does this by adopting representative values for three geometric ratios: the ratio of the clearance between the tubes and holes in the baffles to the tube outside diameter, the ratio of the clearance between the baffle and the shell to the shell inside diameter, and the ratio of the shell inside diameter to the outside diameter of the tube bundle. These ratios are closely related to the various bypass flow streams shown in Figure 5 and are kept as small as possible, consistent with manufacturing tolerances and the clearances required for assembly.

Once these ratios are fixed, Tinker showed that the various bypass flow rates become functions of five more geometric ratios: the ratio of the shell diameter to the tube length between baffles, the ratio of the tube pitch to the tube length between baffles, the ratio of the tube pitch to the tube outside diameter, the ratio of the baffle height to the shell diameter, and the ratio of the number of rows of tubes traversed per pass to the shell inside diameter divided by the tube pitch.

The modifications to the Reynolds number are made using correction factors which depend upon the various clearance ratios previously cited. Once the modified Reynolds number is calculated, the heat transfer coefficient and shell-fluid friction factor can be determined from the appropriate chart.

As one would suspect, Tinker's calculated values, with their many complex relationships, have been proven to be as much as 50 percent deviant from the experimentally determined values. Tinker was however, the forerunner in this area, and his published works have stimulated many advances in this area.[11,12]

E. Devore Method

Anthony Devore used the basic method of Tinker but made several simplifications. First, he used standard tolerances for commercial exchangers in lieu of certain of Tinker's geometric ratios. Second, he limited the number of baffle cut ratios. Lastly, he provided several charts and tables for easy determination of correction factors. The results are even more conservative than those obtained via Tinker's original method.[13]

F. Numerical Methods

The most promising method of calculating the design parameters for a shell and tube heat exchanger stems directly from sophisticated computer-based numerical procedures for calculating fluid-flow distributions. Shell and tube exchanger geometries necessitate a three-dimensional procedure. Three velocity components have to be computed for the shell fluid at each point. Furthermore, the lack of any axial symmetry causes these velocities, as well as the temperatures and

pressures, to vary throughout all three space dimensions. The solution to the differential equations which govern the flow of a continuous fluid can be accomplished by using a finite-difference approach. Alternately, the space within the shell can be regarded as uniformly filled with fluid, through which is distributed, on a fine scale, a resistance to fluid motion.

This alternate continuum approach has two advantages. First, it allows the use of a grid which is less fine than the finite-difference approach, with a savings in computer time. Second, it allows for the use of the extensive experimental data that is available on the pressure-drop and heat transfer performance of fluid flow through tube banks. The results of this type of computer-based numerical method is an increased realism of the predictions of the pressure drop and heat transfer coefficient.[14]

G. Proprietary Programs

During the boom years of heat transfer development, universities and industries worked hand in hand to solve heat transfer problems. Since that time, the emergence of the new fields of nuclear energy and aerospace flight coupled with the revolutionary computational power made available by high-speed digital computers caused a partial disjunction between this university-industry fellowship. Professor Thomas Sherwood from the Massachusetts Institute of Technology describes it in these words:

...Prior to World War II the emphasis at universities was on the collection and correlation of data intended to be of direct use by the practicing design engineer.

Industry had few such data and published little, so schools felt a responsibility to fill the need. This urge to be immediately helpful to industry has largely disappeared today; research in schools is now along more scientific and theoretical lines, hopefully of value to industry a generation hence. Our rapport with industry has suffered.[15]

The new fields of nuclear energy and aerospace flight transfused research funds of unprecedented magnitude into the laboratories and research institutes. The results of this expanded research had, unfortunately, little applicability to industrial needs.

The computer's mammoth capabilities enabled theorists and design engineers alike to solve the heat transfer problems using numerical methods that were abandoned in the boom years due to their computational sophistication. Computer programs were rapidly developed and the need for the university's experimental data ebbed.

What evolved to take the place of university research were organizations that did private research for the benefit of the members of their consortium. Their experimental data along with their computer source-codes were proprietary. The suppression of publicly available information and experimental data was widespread, and today the effects of this lack of scholarly interchange cannot really be discerned.

In the area of heat transfer, one organization in particular stands out as being the most knowledgeable in the field. Heat Transfer Research Incorporated (HTRI) in Alhambra, California, has developed the most sophisticated software in use today in the design of shell and tube

heat exchangers. The majority of the shell and tube heat exchanger manufacturers use their programs, such as ST-4 to compute the heat transfer characteristics and pressure drops in their heat exchangers.[16,17]

V. LIQUID-LIQUID SHELL AND TUBE HEAT EXCHANGER UTILIZATION

Because shell and tube heat exchangers can be constructed with very large heat transfer surfaces in relatively small volume, can be fabricated from alloy metals to resist corrosion and can be used for heating and cooling virtually all fluids, they are universally accepted as the most important class of heat transfer equipment.[18]

From cooling chemical solutions in world-class chemical processing plants to heating water for a backyard swimming pool, applications for liquid-liquid shell and tube heat exchangers abound. The nuclear industry uses them to cool heavy water in atomic reactor installations. The metallurgical industry uses them to cool quench oil. The Food Processing industry uses them for vegetable oil heaters and dairy product coolers. The petroleum industry uses them to cool oil used in machine lubrication on offshore oil platforms by using sea water as the cooling medium. Local schools may use them to preheat low-grade fuel oil. The list of applications will be as long as the imagination of the design and process engineer is broad.[19,20]

Virtually all industries that produce a waste or by-product fluid stream that has an exit temperature higher than the inlet fluid temperature can use a liquid-liquid shell and tube heat exchanger to recover some of the thermal energy that would otherwise be lost. Particularly during the present period of high energy costs, the capital outlay for a heat exchanger is recouperated via lowered heating costs allowed by the thermal energy retrieved from the waste stream.

VI. PROBLEMS ENCOUNTERED IN SHELL AND TUBE HEAT EXCHANGER DESIGN

A. General

The designer of shell and tube heat exchangers is plagued with an enormous amount of uncertainties. Two important problems that must be dealt with, that are particularly troublesome in shell and tube heat exchangers designed for liquid-liquid applications are flow induced vibrations and fouling. Each of these problems will be discussed and possible solutions will be provided.

B. Flow Induced Vibrations

The cross-flow component of the shell-side fluid stream has been held primarily responsible for the widespread failure of tubes that occurred in the late 1960's. Other factors, such as baffle configurations and unsupported tube spans, were also defined as having at least a part in the destructive vibratory action. Present-day understanding of this complex phenomenon is limited. In fact, TEMA standards allow a disclaimer that the manufacturer is not responsible or liable for any direct, indirect, or consequential damages resulting from vibration.[21]

Mechanical failure due to flow-induced vibrations can occur in several different ways. The amplitude of the vibration may be large enough that two tubes collide with each other or a peripheral tube will strike the shell inner wall. In either case, the tube wall is worn down, and the tube eventually leaks. The tube may repeatedly impact with the inside of the baffle hole causing the same type of abrasion. Failure can also be caused by vibration-induced stress fluctuations in the tube when macroscopic flaws on the tube surface propagate to serious

cracks.[22] In any case, the result is the same. The heat exchanger's efficiency is degraded and eventually must be shut down for costly repair or retubing. Without necessary design modifications however, the repaired heat exchanger will be put back into service and will continue to have serious problems caused by the unwanted vibrations.

Presently, world-wide research into this problem has produced only a few experimentally proven methods to reduce vibrations. Since field experience has shown that tubes in the area of the baffle cuts are the most prone to vibrate, the utilization of the "no tubes in window" baffle configuration shown previously in Figure 3 obviously eliminates these vibration-prone tubes. The draw-back here is that in order to accommodate the reduced tube area, the shell diameter must be increased. Another method widely acclaimed is the insertion of helical rods as tube spacers at the midpoint of unsupported tube spans. Stability is added and vibration is reduced, but the trade-off is flow diversion and consequent increased fluid friction caused by these obstructions. Simple remedies such as decreasing the hardness of the baffle material and increasing the lightness of the tube-to-tube bundle fit have had limited success. No doubt, the present day research in this area will lead to a better understanding of this phenomenon and result in newer methods of design which will reduce or eliminate the likelihood of vibration.[23]

C. Fouling

Undesirable buildup of deposits on tubes have plagued shell and tube heat exchanger manufacturers and users since the first unit was placed into operation. Performance degradation of up to 80%, and complete failures were commonplace. The mechanisms of fouling have been

categorically identified as: corrosion fouling, precipitation fouling, particulate fouling, chemical reaction fouling and biological fouling. The identification of the mechanisms was relatively easy, the difficulty has been the development of empirical models to characterize and predict each of the mechanisms. In spite of enormous research in this area, good prediction methodology is lacking. Designers still add typical fouling resistances based on past experience to their calculations in determining the overall heat transfer coefficient for the exchanger. This intentional oversizing of the exchanger considerably increases the capital cost of the exchanger, yet does nothing to deter the fouling from taking place. In fact, this design practice can lead to catastrophic results, in that a unit designed as a sensible heat exchanger may initially operate as a boiler.[24,25]

Due to our lack of understanding of fouling, techniques to reduce its damaging effects are limited. The basic rule in design for fouling is to place the fluid which causes the greatest amount of fouling inside the tubes. There, stagnant regions, which cause the worst fouling, are less likely to occur. Furthermore, when the inevitable maintenance due to fouling comes about, the mechanical cleaning of the insides of the tubes is usually much easier than removing the scale on the outside of tubes in a tube bundle.[26] The use of exotic metals such as zirconium and titanium in the manufacture of the tubes has proven to be effective in decreasing the rate of fouling build-up.[27] Finally, the Heresite-Saekaphen Company in Wisconsin has patented a tube coating system which prevents fouling, and is resistant to erosion, corrosive chemicals and solvents. Experimental and real-life results have shown

that the service life of tube bundles has increased more than five-fold. The cost for this .006 inch phenolformaldehyde coating is cost-effective since the coating eliminates the need to clean the tubes, and allows for an initial exchanger design that is not oversized.[28] Just as with flow induced vibrations, the ongoing research in this area will bring about new innovative methods to combat the fouling problem.

VII. COMPUTER-AIDED DESIGN PROCEDURE

A. General

In order to provide prospective shell and tube heat exchanger buyers a preliminary size and cost of a shell and tube heat exchanger that will accomplish the required sensible heat transfer, a computer program was developed. The program designs exchangers with TEMA Type E shells from six to sixty inches in two inch increments. One shell-side pass is provided with either two, four, six, or eight tube-side passes. The program is limited to liquid-to-liquid applications where pressures are less than 300 pounds per square inch.

The program is user-friendly in that instructions for the use of the program are initially displayed, and the user needs to respond only to questions asked. Examples of input are provided, and required units are clearly established. The input data can be modified and final design configurations can be compared. The nomenclature used throughout this chapter can be found in Appendix A. The main program source code is found in Appendix B.

B. Instructions

The initial instructions include remarks about the program's capabilities, limitations, and required input data. The required inputs are: tube-fluid entrance temperature (TT1), tube-fluid exit temperature (TT2), shell-fluid entrance temperature (TS1), tube-fluid flow rate (MT), shell-fluid flow rate (MS), maximum tube length (LRESTR), maximum allowable shell-side pressure loss (AMAXSP), and maximum allowable tube-side pressure loss (AMAXTP). The temperature units are in °F, the flow

rates are in gallons per minute, lengths are in feet and pressure losses are in pounds per square inch. The initial instructions also provide a listing of the major works that were used as sources for the program's development.

C. Input Data

The user is asked to make a choice of tubing to be used in the exchanger from a listing of thirty-six common tube sizes and Birmingham Wire Gages (BWG). Tube outside diameters (DTOT) range from 0.75 to 1.5 inches and the range of BWG is from seven to twenty. A choice of tube material must also be made from a selection of eighteen different metals and alloys ranging from common carbon steel to the more exotic metals of titanium and zirconium.

The next step is to choose the fluids that are to be used. Seven fluids are available for selection, they are: water, methyl alcohol, gasoline, Dowtherm A, kerosene, SAE 10 lubricating oil, and ethylene glycol. In keeping with design procedures, the fluid that will cause the worst fouling should be placed inside the tubes. Any combination of these seven fluids are design possibilities.

The input data previously discussed is input to the computer as prompted for by the program. The last required input is whether or not the addition of fouling factors is desired to be used in the calculations. A negative response to this question will result in the heat exchanger being sized utilizing the overall heat transfer coefficient calculated assuming clean tubes. This may result in the heat exchanger not meeting its design performance sometime in the future if no precautions are taken to guard against fouling.

exchanger flow arrangement. These parameters are useful in allowing for compact graphical presentation of exchanger performance. Figure 8 graphically shows the relationship between these parameters for a heat exchanger with one shell pass and 2-, 4-, 6-, or 8-tube passes. The algebraic relationship between the number of transfer units, the capacity rate ratio and the heat exchanger effectiveness is:[39]

$$NTU = -(1+C^2)^{-1/2} \ln \left[\frac{\frac{2}{E} - 1 - C - (1+C^2)^{1/2}}{\frac{2}{E} - 1 + C + (1+C^2)^{1/2}} \right]$$

$$\text{where } C = \frac{C_{MIN}}{C_{MAX}}$$

Although this equation was derived for a heat exchanger with one shell pass and two tube passes, it is used as the basic NTU-Relationship. The equations that result from the study of 4-, 6-, or 8-pass exchangers yield results that are numerically so close to the two-tube-pass situation that nothing is to be gained by the utilization of these more complex equations.[40]

M. Calculation of Estimated Overall U-Value

Tables of U-values typical of various applications of shell and tube exchangers are widespread. They normally are tabulated giving a range of U-values that can be expected for two particular fluid streams. For example, a water-to-water shell and tube heat exchanger will have a U-value of between 250 - 300 BTU/hr-ft²-°F. This is in contrast to a water-to-heavy organic liquid shell and tube heat exchanger which would

The fluid with the lowest capacity rate is called the minimum fluid (CMIN). The fluid with the greatest capacity rate is called the maximum fluid (CMAX). The capacity rate ratio is defined as CMIN/CMAX.

K. Heat Transfer Effectiveness Calculation

Heat exchanger effectiveness is defined as the ratio of the actual heat transfer to the maximum possible heat transfer. Kays and London use this equation for effectiveness:[37]

$$\epsilon = \frac{q}{q_{\max}} = \frac{C_h(t_{h,\text{in}} - t_{h,\text{out}})}{C_{\min}(t_{h,\text{in}} - t_{c,\text{in}})} = \frac{C_c(t_{c,\text{out}} - t_{c,\text{in}})}{C_{\min}(t_{h,\text{in}} - t_{c,\text{in}})}$$

where C_{\min} is the smaller of the C_h and C_c magnitudes.

The thermodynamically limited maximum heat transfer rate (q_{\max}), would only be realized in a counterflow heat exchanger of infinite heat transfer area. Although this effectiveness is in reality a heat transfer effectiveness, the above equation reduces to a temperature ratio. Using nomenclature from Appendix A:

$$\text{EFF} = \frac{|T_{2\text{MIN}} - T_{1\text{MIN}}|}{|T_{1\text{MAX}} - T_{1\text{MIN}}|}$$

L. Determination of the Number of Transfer Units

The number of heat transfer units (NTU) is a nondimensional expression of the heat transfer size of the exchanger. Kays and London use this equation for number or exchanger heat transfer units:[38]

$$N_{tu} = \frac{AU_{av}}{C_{\min}} = \frac{1}{C_{\min}} \int_0^A U \, dA$$

The number of exchanger heat transfer units is a function of three parameters: the effectiveness, the capacity rate ratio, and the heat

In Case 2:

$$TS2 = TS1 - \frac{Q \times 7.4805}{MS \times DENS \times CPS \times 60.0}$$

Of course, the fluid properties used in the previous two equations were only assumed average values. Once the initial value for the shell exit temperature is obtained, the mean fluid temperature is calculated by:

$$TSAVE = \frac{TS1 + TS2}{2}$$

Using this value for the mean shell-fluid temperature, the fluid property subroutines are called, the corrected values of density and specific heat are obtained. A new value for the shell-fluid exit temperature is calculated using the aforementioned equations, and the new value for the shell-fluid exit temperature is compared to the previously calculated value. If they differ by more than one degree of temperature the iterative process is continued. If the two values for the shell-fluid exit temperature do not differ by more than a single degree, the last calculated value for the exit temperature is recognized as the real shell-fluid exit temperature for use in any future calculations.

J. Calculation of Maximum and Minimum Capacity Rates

The mass flow rate x the specific heat of a fluid is defined as the capacity rate of that fluid.[36] Equations to calculate the capacity rates of the shell and tube fluid are:

$$CS = \frac{MS \times DENS \times CPS \times 60.0}{7.4805}$$

$$CT = \frac{MT \times DENT \times CPT \times 60.0}{7.4805}$$

I. Shell-Fluid Exit Temperature Determination

Since the exit temperature shell-fluid is an unknown, it can be found by doing an energy balance on the heat exchanger. Two possible conditions could exist. In Case 1, the tube fluid is being heated. Here the energy gained by the tube fluid must be lost by the shell-fluid. In Case 2, the tube-fluid is being cooled. Here the energy lost by the tube-fluid must be gained by the shell-fluid. The heat transfer rate of the tube-fluid is dependent upon these factors: the tube-fluid flow rate, the tube-fluid density, the tube-fluid specific heat, and the magnitude of the tube-fluid exit and entrance temperature difference.

In Case 1:

$$Q = \frac{MT \times DENT \times CPT \times 60.0}{7.4805} \times (TT2 - TT1)$$

In Case 2:

$$Q = \frac{MT \times DENT \times CPT \times 60.0}{7.4805} \times (TT1 - TT2)$$

Note that 60.0 is a conversion factor for minutes per hour and 7.4805 is a conversion factor for gallons per cubic feet.

Since an average shell-fluid temperature cannot be determined until the exit temperature of the shell-fluid is known, average values for the fluid properties are assumed depending upon the shell-fluid choice.

An iterative procedure will be followed in order to obtain the correct value of the shell exit temperature. The equations are:

In Case 1:

$$TS2 = TS1 - \frac{Q \times 7.4805}{MS \times DENS \times CPS \times 60.0}$$

H. Thermo-Hydraulic Properties of the Tube Fluid

Four basic properties of each fluid must be known to design a liquid-to-liquid heat exchanger. These are: density, viscosity, specific heat and thermal conductivity. The temperature at which the fluid properties are determined is the mean fluid temperature, which is simply the arithmetic average of the entrance and exit temperatures. The mean tube fluid temperature is:

$$TTAVE = \frac{(TT1 + TT2)}{2}$$

This computer program calculates these four fluid properties by way of calling separate subroutines. Each of the seven fluid choices has four subroutines which are associated with it, one for each of the fluid properties. For example, water has these subroutines:

| <u>Subroutine</u> | <u>Desired Fluid Property</u> |
|-------------------|-------------------------------|
| DWATER | Density |
| CPWATR | Specific Heat |
| VWATER | Dynamic Viscosity |
| KWATER | Thermal Conductivity |

Each subroutine contains temperature dependent data of its particular fluid property. An interpolation algorithm is employed to calculate the desired fluid temperature if the mean fluid temperature does not match the temperature data points filed for that particular fluid property. A complete listing of all of the fluid property subroutines for water is found in Appendix C. The other fluid subroutines are strikingly similar in form to these subroutines.

between the tubes, which depends on the tube pitch and tube size, that make up the crossflow area.

The tube clearance is simply:

$$TC = TPH - DTOT$$

The calculations of the shell-side flow rates can now be inscribed. The minimum shell-side flow rate for a heat exchanger using a K1 baffle spacing is:

$$QSK1L = 3.116875 \times (0.2 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 2 \text{ ft/sec}$$

The maximum shell-side flow rate for the same exchanger is:

$$QSK1H = 3.116875 \times (0.2 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 4.5 \text{ ft/sec}$$

The flow rate equations for exchangers which use the other two baffle spacings are:

For a K2 baffle spacing:

$$QSK2L = 3.116875 \times (0.45 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 2 \text{ ft/sec}$$

$$QSK2H = 3.116875 \times (0.45 \times DSI) \times \frac{DSIU}{TPH} \times TC \times 4.5 \text{ ft/sec}$$

For a K3 baffle spacing:

$$QSK3L = 3.116875 \times DSI \times \frac{DSIU}{TPH} \times TC \times 2 \text{ ft/sec}$$

$$QSK3H = 3.116875 \times DSI \times \frac{DSIU}{TPH} \times TC \times 4.5 \text{ ft/sec}$$

G. Maximum and Minimum Shell-Side Flow Rates

In keeping with generally recommended practice the shell-side velocity should be at least 2 ft/sec but not more than 4.5 ft/sec.[33,34] The shell-side crossflow area is dependent on three factors: the tube pitch, the diameter of the shell, and the baffle spacing. The only parameter that has not been discussed previously is the baffle spacing. The baffle spacing is the distance between baffles within the exchanger. There is a generally recommended range of baffle spacings. The baffles should be no closer than a distance equal to one-fifth the shell diameter and should not be spaced farther apart than a distance equal to the shell diameter.[35] In keeping with this practice, three shell baffle spacings are considered in the calculations. They are the two extreme baffle spacing conditions and one midway between these two extremes. Numerically the baffle spacings are defined:

| <u>Baffle Spacing</u> | <u>Calculation</u> |
|-----------------------|--------------------|
| K1 | $DSI \times 0.2$ |
| K2 | $DSI \times 0.45$ |
| K3 | $DSI \times 1.00$ |

The estimation of the shell-side cross flow area follows a calculation procedure that is based upon the previously cited work of Kern.

If the shell inside diameter is divided by the tube pitch, a fictitious number of tubes which may be assumed to exist in the center of the shell is obtained. In actuality, there is more than likely no central row of tubes but rather a row of tubes on either side of the center with less tubes than is calculated by this procedure. This departure from reality is neglected. For each tube there is considered to be a clearance

$$QTH8 = \frac{3.116875 \times NT8 \times ATIN \times 8.0 \text{ ft/sec}}{8.0 \text{ passes}}$$

These two equations incorporate the assumption that there is an equal amount of tubes in each of the eight tube passes. This is consistent with actual shell and tube exchanger manufacture.[32]

The calculations for the remainder of the construction configurations are quite similar. For 6-pass construction:

$$QTL6 = \frac{3.116875 \times NT6 \times ATIN \times 2.0 \text{ ft/sec}}{6.0 \text{ passes}}$$

$$QTH6 = \frac{3.16875 \times NT6 \times ATIN \times 8.0 \text{ ft/sec}}{6.0 \text{ passes}}$$

For 4-pass construction:

$$QTL4 = \frac{3.116875 \times NT4 \times ATIN \times 2.0 \text{ ft/sec}}{4.0 \text{ passes}}$$

$$QTH4 = \frac{3.116875 \times NT4 \times ATIN \times 8.0 \text{ ft/sec}}{4.0 \text{ passes}}$$

For 2-pass construction:

$$QTL2 = \frac{3.116875 \times NT2 \times ATIN \times 2.0 \text{ ft/sec}}{2.0 \text{ passes}}$$

$$QTH2 = \frac{3.116875 \times NT2 \times ATIN \times 8.0 \text{ ft/sec}}{2.0 \text{ passes}}$$

$$NT6 = ARAV \times F1 \times F6$$

$$NT8 = ARAV \times F1 \times F8$$

E. Available Tube Surface Area

A useful preliminary area calculation is how much heat transfer area is provided per foot of length for a given tube bundle. The external surface area of a tube (ATSU) is fixed once the size is picked, so the area calculations for each of the construction configurations are:

$$AL2 = NT2 \times ATSU$$

$$AL4 = NT4 \times ATSU$$

$$AL6 = NT6 \times ATSU$$

$$AL8 = NT8 \times ATSU$$

F. Maximum and Minimum Tube-Side Flow Rates

In keeping with generally recommended practice the tube-side velocity should be at least 2 ft/sec but not more than 8 ft/sec.[30,31] With this in mind, the maximum and minimum flow rates for 2-, 4-, 6-, and 8-pass construction can be calculated. The tube inside area was fixed with the selection of the size of the tubing. To keep units correct, a conversion factor of 3.116875 is used. This allows for the flow rate to be in gal/min while the velocity is in ft/sec and the area is in square inches. The minimum flow rate for 8-pass construction is:

$$QTL8 = \frac{3.116875 \times NT8 \times ATIN \times 2.0 \text{ ft/sec}}{8.0 \text{ passes}}$$

The maximum flow rate for 8-pass construction is:

$$NT2 = F1 \times F2 \times ARAV$$

where $F1$ is a correction factor which accounts for the tube layout design. For triangular tube layouts, $F1 = 0.9069$. $F2$ is a correction factor which accounts for the loss of tubes due to pass partition dead space in 2-pass construction.

$$F2 = 1.0 - \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

The equations for the number of tubes in a shell with 4-, 6-, or 8-pass constructions are the same as for the 2-pass construction except that the $F2$ correction factor is changed due to the increased amount of dead space caused by the increase in the necessary flow partitions. The correction factors for these construction types are:

$$F4 = 1.0 - 2.125 \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

$$F6 = 1.0 - 3.25 \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

$$F8 = 1.0 - 4.0 \left(\frac{TH \times DSIU}{\pi/4 \times DSIU^2} \right)$$

and the corresponding equations for calculating the number of tubes for these construction types are:

$$NT4 = ARAV \times F1 \times F4$$

The number of tube-side passes affects the number of tubes in the shell. In Figure 2, partitions designs were shown. It is clear that no tubes may be placed in the plane of a partition. The thickness (TH) of this partition is dependent upon the size of the shell as follows:

| <u>Range of DSI(in)</u> | <u>TH(in)</u> |
|-------------------------|---------------|
| 6.0 to 18.0 | 0.25 |
| 20.0 to 36.0 | 0.50 |
| 38.0 to 60.0 | 0.75 |

The larger the number of tube-side passes, the smaller the number of tubes in the shell.

The size of the shell has an obvious part in the determining the number of tubes in the exchanger. The larger the shell diameter, the greater the number of tubes if all other parameters were kept constant. The entire shell diameter is not completely usable however. Structural necessity requires that at least a one-quarter inch free-space at the end of the tube sheet be maintained. Therefore the usable diameter (DSIU) calculation is as follows:

$$DSIU = DSI - 0.5$$

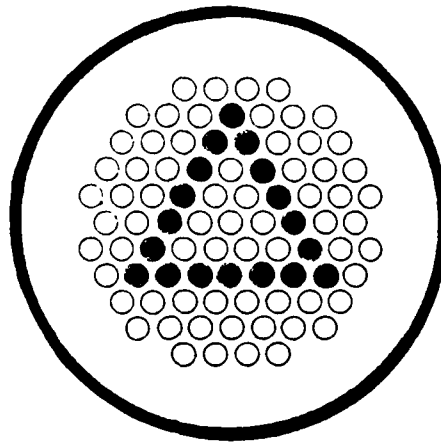
The maximum number of tubes that are possible in a given shell is calculated from:

$$ARAV = \left(\frac{DSIU - DTOT}{TPH} \right)^2$$

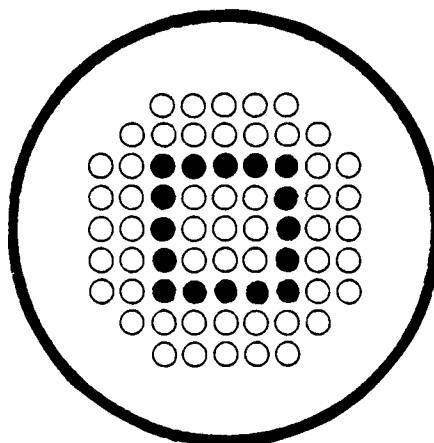
The actual number of tubes that will be found in a given shell with two tube-passes is given by:[29]

FIGURE 7

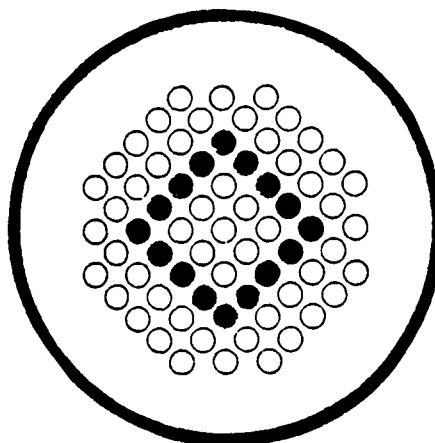
STANDARD TUBE LAYOUT DESIGNS



TRIANGULAR TUBE LAYOUT



SQUARE TUBE LAYOUT



ROTATED SQUARE TUBE LAYOUT

D. Number of Tubes Calculations

The number of tubes possible in a given shell diameter is dependent upon five factors: the tube size, the tube layout design, the tube pitch (TPH), the number of tube-side passes, and the diameter of the shell (DSI). Each of these parameters will be explained and the final equation for calculating the number of tubes will be presented.

The tube size fixes the outside diameter of the tubes. All other parameters equal, it is obvious that more 1.0 inch tubes would fit in a given shell than would 1.5 inch tubes.

The tube layout design is the pattern or tube construction. Three patterns have widespread usage today, they are: the triangular pattern, the square pattern and the oriented square pattern. Each of these patterns are shown in Figure 7. The spacing is clearly different for the triangular and square patterns. This computer design uses calculations for triangular tube layouts only.

The holes in the tubesheet cannot be drilled very closely together, since too thin a separation structurally weakens the tubesheet. The shortest distance between two adjacent tube holes is the clearance which has become standard in practice. The tube pitch is the shortest center-to-center distance between adjacent tubes. This dimension, which depends on the size of the tubing used, is also standard. For triangular tube layouts, the following tube pitches have been designated:

| <u>DTOT(in)</u> | <u>TPH(in)</u> |
|-----------------|------------------|
| 0.75 | 15/16 or 0.9375 |
| 1.00 | 1 1/4 or 1.25 |
| 1.25 | 1 9/16 or 1.5625 |
| 1.50 | 1 7/8 or 1.875 |

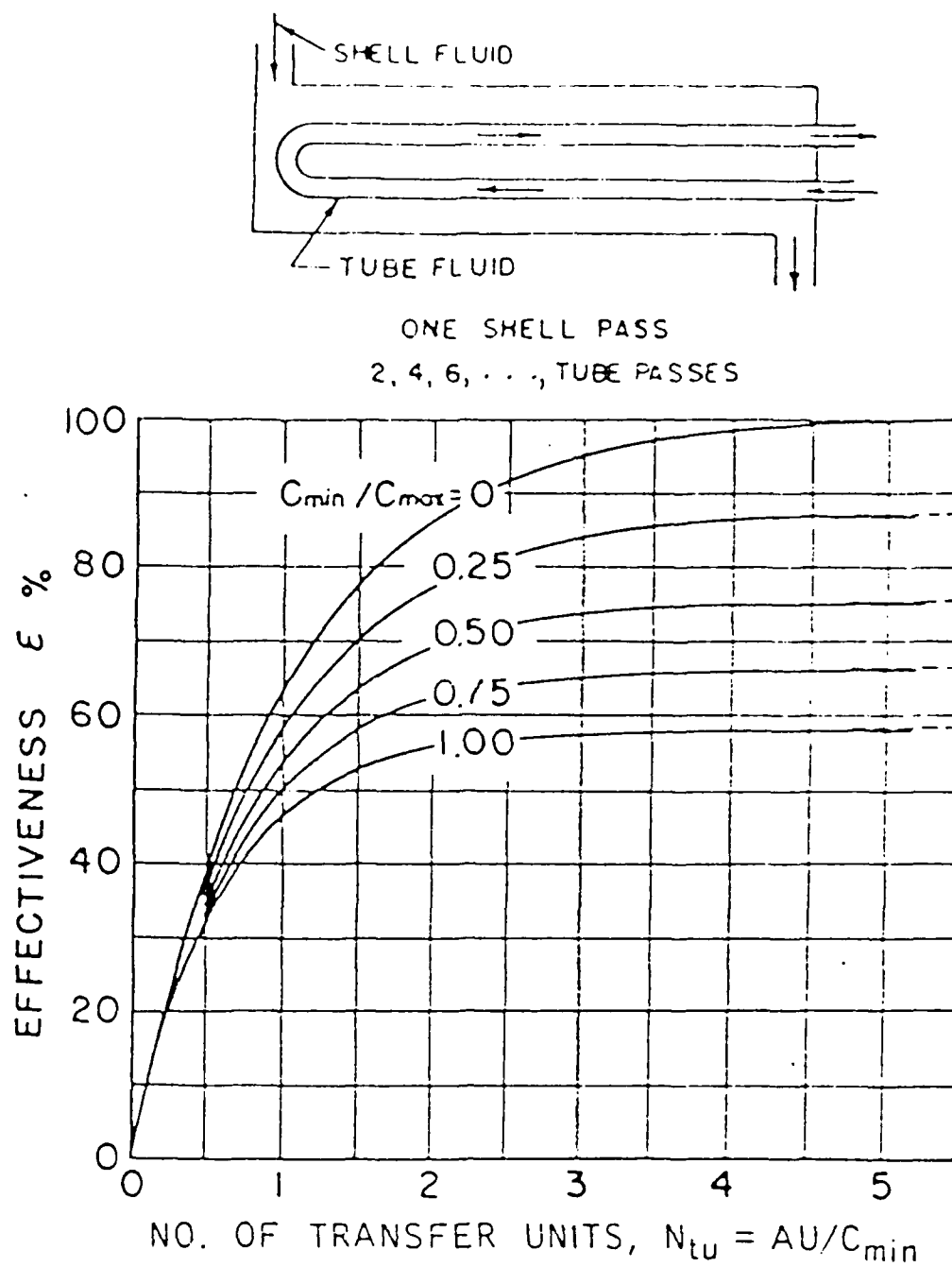
At this point, all the data that the computer program needs to provide a list of shell and tube heat exchanger configurations that will accomplish the required heat transfer is provided except for any restrictions that may need to be imposed.

The site that the heat exchanger will eventually occupy may be fixed, and only a certain space may be available. The length of the heat exchanger would certainly need to be restricted. Moreover, as the heat exchanger would most probably need some sort of inspection or maintenance done to it during its lifetime, it is common practice to require that the maximum length of the heat exchanger be less than one-half the available free-space at the site. This is because maintenance of the tubes requires the removal of the shell and obviously one would need at least twice the length of the heat exchanger to accomplish this task. Therefore, the user is asked to input the maximum allowable length of the heat exchanger to the program.

Another important limitation that normally is imposed on shell and tube heat exchanger designers is the amount of pressure drop that is allowed through the tubes and the shell. Normally, the size of the pumps and corresponding pumping power is fixed, and therefore a limit on the maximum head loss through a unit must be kept within known limits. Even if the pump sizes are not predetermined, reasonable pressure loss restrictions are imposed since the larger the pressure loss allowed, the larger the pumping power required. This capital cost outlay must be considered in the decision for optimum design. The user inputs these maximum allowable pressure losses as prompted by the program.

FIGURE 8

EFFECTIVENESS - NUMBER OF TRANSFER UNITS RELATIONSHIP FOR A 1-2
SHELL AND TUBE HEAT EXCHANGER



have a U-value of between 40-75 BTU/hr-ft²-°F. These values are for unfouled tubes. Average (mid-range) values for all possible combinations of the seven fluid choices have been incorporated into the computer program as a design check of the actual U-value which is calculated later in the program. The source used for these estimated U-values is Heat Exchangers by Kakac and Bergles. These values are for plain (unfinned) tubes.

Tabulated values of fouling resistances are also common. If the user wants a fouling factor (RFOULT) added to the calculation of the overall U-value, the following equation is used.

$$U = \frac{1}{R_{FOULT} + \frac{1}{U_C}}$$

where U_C is the U-value for clean (unfouled) tubes. Table 1 lists the overall U-values and fouling resistances used in the program.[41]

N. Determination of Possible Exchanger Configurations

Once the tube-fluid flow rate, the shell-fluid flow rate, the maximum tube length, and the maximum allowable tube-side and shell-side pressure losses have been specified, the computer does a systematic check of every possible heat exchanger configuration to see if it meets the problem specifications. It checks each available shell size (from six to sixty inches), with every possible number of tube-side passes (2, 4, 6, or 8) and every possible baffle spacing (0.2, 0.45, and 1.0), a total of 336 possible exchanger configurations, to see which, if any, meet the problem requirements.

TABLE 1
TYPICAL OVERALL DESIGN COEFFICIENTS FOR
SHELL AND TUBE HEAT EXCHANGERS

| <u>FLUID 1</u> | <u>FLUID 2</u> | TOTAL FOULING RESISTANCE HR-FT ² -°F | OVERALL U-VALUE BTU |
|---------------------------|---------------------------|--|----------------------------|
| | | <u>BTU</u> | <u>HR-FT²°F</u> |
| Water | Water | 0.0015 | 275.0 |
| Water | Light Organic Liquids | 0.0015 | 150.0 |
| Water | Medium Organic Liquids | 0.002 | 100.0 |
| Water | Heavy Organic Liquids | 0.0025 | 57.5 |
| Light Organic Liquids | Light Organic Liquids | 0.002 | 115.0 |
| Light Organic Liquids | Medium Organic Liquids | 0.0025 | 85.0 |
| Light Organic Liquids | Heavy Organic Liquids | 0.003 | 50.0 |
| Medium Organic Liquids | Medium Organic Liquids | 0.003 | 65.0 |
| Medium Organic Liquids | Heavy Organic Liquids | 0.0035 | 35.0 |
| Heavy Organic Liquids | Heavy Organic Liquids | 0.005 | 20.0 |

Note: Light Organics include: Methyl Alcohol and Gasoline

Medium Organics include: Dowtherm A and Kerosene

Heavy Organics include: SAE 10 Lubricating Oil and Ethylene Glycol

The first check that is made is to see whether the tube-fluid flow rate falls within the maximum and minimum allowable tube-fluid flow rates for that particular pass configuration. If the tube-fluid flow rate is acceptable, the shell-fluid flow rate is checked to see if it falls within the range of maximum and minimum allowable shell-fluid flow rates for any of the three baffle spacings. Both the tube-fluid and the shell-fluid must fall within their respective acceptance ranges for further checks to continue.

If the tube-fluid and shell-fluid flow rates are acceptable a pressure loss and length check subroutine PRESCK, is called. A listing of PRESCK is found in Appendix D.

This subroutine contains the bulk of the thermal-hydraulic design equations. Each of the major equations and their source will be explained in detail.

The velocities of the tube-fluid and the shell-fluid are calculated. The shell-fluid velocity (VS) is simply the shell-fluid flow rate (MS) divided by the crossflow area of the shell (ASC). The evaluation of the shell crossflow area was previously discussed in the determination of the minimum and maximum shell-side flow rates of the exchanger. The shell-fluid velocity reduces to:

$$VS = \frac{MS}{ASC} \times C$$

where C is a conversion factor (= .3208342) which allows velocity to be in ft/sec while mass flow rate is input in GPM and the area is in in².

The tube-side fluid velocity (VT) is simply the tube-fluid flow rate (MT) divided by the total tube-side area per pass. As noted earlier,

the number of tubes that will be found in a given heat exchanger will decrease as the number of tube-passes increases. The tube-side velocity equation will therefore differ for each pass-construction type. This is because the tube-side area is equal to the number of tubes multiplied by the internal area of the tubes (ATIN). The tube-side velocity equation is for 2-pass construction is:

$$V_T = \frac{C \times M_T}{ATIN \times \frac{NT^2}{2.0}}$$

where C is the same conversion factor that was used in the calculation of the shell-side velocity. The equations for 4-, 6-, or 8-pass construction are the same with the exception of the substitution of the appropriate number of tubes per pass for that particular pass construction.

The mass velocity of the shell-fluid (GS), and the mass velocity of the tube-fluid (GT) can now be found by multiplying the velocities by the density of the particular fluid.

$$GS = VS \times DENS$$

$$GT = VT \times DENT$$

The next step is to calculate the dimensionless Reynolds Number. The calculation of the tube-side Reynolds Number (RET) is straightforward. This is simply the tube-side mass velocity multiplied by the tube inside diameter (DTIN) divided by the tube-fluid dynamic viscosity (DVT).

$$RET = \frac{\frac{DTIN}{12.0} \times GT}{DVT}$$

The tube inside diameter of the tube, which is in inches, is divided by the conversion factor 12.0 to get it into proper units of feet.

The calculation of the shell-side Reynolds Number (RES) is not as direct. There is not a circular flow channel for the shell-fluid. An equivalent diameter must be calculated. The direction of the shell-fluid flow is partially along and partially at right angles to the long axes of the tubes in the tube bundle. The flow area at right angles to the long axes varies with each row of tubes. A hydraulic radius based upon the flow area across any one row could not distinguish between standard tube layout designs. In order to obtain a simple correlation combining both the tube size, the tube pitch (TPH), and their type of layout, excellent agreement is obtained if the hydraulic radius is calculated parallel to instead of perpendicular to the long axes of the tubes. The equivalent diameter for the shell is taken as four times the hydraulic radius obtained for that particular tube layout design. For the triangular tube layout design the shell equivalent diameter (DEQS) is:[42]

$$DEQS = \frac{4 \times \left(\frac{TPH}{2} \times 0.86 TPH - \frac{\pi DT0^2}{8} \right)}{\frac{\pi DT0}{2}}$$

where DT0 is the outside diameter of the tubes.

The evaluation of the shell-side Reynolds Number is now found by:

$$RES = \frac{\frac{DEQ}{12.0} GS}{DVS}$$

where DVS is the dynamic viscosity of the shell-fluid.

Evaluation of the shell-side heat transfer film coefficient (H0) can be done using this equation:[43]

$$\frac{h_{de}}{k} = 0.36 (Re)^{0.55} (Pr)^{1/3} \left(\frac{\mu}{\mu_f} \right)^{0.14}$$

using Appendix A nomenclature:

$$H0 = \frac{\frac{KS}{DEQS}}{12.0} 0.36 RES^{0.55} \left(\frac{CPS \times DVS \times 3600.0}{KS} \right)^{1/3} \left(\frac{DVS}{DVSF} \right)^{0.14}$$

where KS is the thermal conductivity of the shell-fluid, CPS is the specific heat of the shell-fluid and DVSF is the dynamic viscosity of the shell-fluid evaluated at the fluid-tube wall interface temperature. This temperature has been estimated to be the average value of the shell-fluid and tube-fluid temperatures. The number 3600.0 in the Prandtl Number term is conversion factor for seconds per minute.

Evaluation of the tube-side heat transfer film coefficient (HI) can be made using the following equation:[44]

$$\frac{h_{id}}{k} = 0.027 (Re)^{0.8} (Pr)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

using Appendix A nomenclature:

$$HI = \frac{KT}{DTIN} \cdot 0.027 \cdot RET^{0.8} \cdot \left(\frac{CPT \times DVT \times 3600.0}{KT} \right)^{1/3} \cdot \left(\frac{DVT}{DVTF} \right)^{0.14}$$

12.0

where KT is the thermal conductivity of the tube-fluid, CPT is the specific heat of the tube-fluid and DVTF is the dynamic viscosity of the tube-fluid evaluated at the fluid-tube wall interface temperature.

The tube wall resistance (RW) is evaluated. Although this resistance can be neglected for many metals, it is evaluated for all since the more exotic metals such as zirconium and titanium have large resistances to heat transfer compared to copper and aluminum. TEMA standards handbook provides this equation for tube wall resistance calculations:[45]

$$RW = \frac{d}{24K} \times \left[\ln \left(\frac{d}{d-2t} \right) \right]$$

using Appendix A nomenclature:

$$RW = \frac{DTOT}{24 KW} \cdot \ln \left[\frac{DTOT}{DTOT - 2(DTOT - DTIN)} \right]$$

where KW is the thermal conductivity of the tube metal which was fixed once the choice of the tube metals was made in the program.

All the parameters needed for the determination of the overall heat transfer coefficient (UCALC) have now been determined. Using another equation from the TEMA standards handbook the U-value is calculated by:[46]

$$U = \frac{1}{\frac{1}{h_o} + r_o + r_w + r_i \left(\frac{A_o}{A_i} \right) + \frac{1}{h_i} \left(\frac{A_o}{A_i} \right)}$$

where r_o is the fouling resistance on the outside of the tubes, r_i is the fouling resistance on the inside of the tubes and $(\frac{A_o}{A_i})$ is the ratio of the outside to inside surface of the tubing. The calculation of the U-value differs depending upon the user's desire to add fouling factors. If the user wants fouling factors added the equation is:

$$U_{CALC} = \frac{1}{\frac{1}{H_O} + R_O + R_W + (R_I \times TRATIO) + (\frac{1}{H_I} \times TRATIO)}$$

where TRATIO is the area ratio described above. R_O and R_I are the fouling resistances on the outside and inside of the tube respectively. They have been computed depending upon the tube-fluid and shell-fluid choices. The total of the R_I and R_O is equal to the total fouling resistances found in Table 1.

If the user wants the U-value calculated for clean tubes the equation simplifies to:

$$U_{CALC} = \frac{1}{\frac{1}{H_O} + R_W + (\frac{1}{H_I} \times TRATIO)}$$

The required heat transfer area ($ACORR$) can now be calculated by:

$$ACORR = \frac{NTU \times C_{MIN}}{U_{CALC}}$$

Using this value of the heat transfer area, the required length of tubing (L) can be calculated depending upon the pass construction. For 2-pass construction:

$$L = \frac{ACORR}{AL2}$$

where AL2 is the tube bundle surface area per foot of tube length for 2-pass construction. The length calculation for 4-, 6-, or 8-pass construction is similar to this equation.

This calculated length must be less than the length restriction imposed by the user. This is the first of three checks that are done in the subroutine PRESCK. All three must be successfully achieved before the exchanger configuration is printed as a possible solution to the heat exchanger problem.

Checks 2 and 3 deal with the maximum allowable pressure losses in the exchanger.

The tube-side pressure loss is calculated using two equations from Kern. The first equation accounts for the pressure drop in tubes. This equation is:[47]

$$\Delta P_t = \frac{f G t^2 L m}{2g \rho D \left(\frac{\mu}{\mu_w} \right)^{0.14}}$$

In Appendix A terminology:

$$PLTF = \frac{FRT GT^2 NP L .006944}{\left(\frac{DTIN}{12.0} \right)^2 AG \left(\frac{DVT}{DVTF} \right)^{0.14} DENT}$$

where NP is the number of the tube passes, .006944 is a conversion factor for pounds per square inch, AG is a gravitational acceleration constant and FRT is a tube-side friction factor. This friction factor

is calculated by a subroutine called TFRICT. A listing of this subroutine can be found in Appendix E. This subroutine contains data on friction factors correlated by Sieder and Tate for fluids which are being heated or cooled in tubes. The subroutine uses a linear interpolation algorithm to calculate the friction factor depending on the Reynolds Number of the tube-side fluid. The Sieder and Tate correlated data was accepted and published by TEMA for use in pressure loss calculations.

Since the fluid must flow from one pass into the next pass by doing a 180° direction change in the tube channel area of the exchanger, an additional pressure drop is experienced. This loss is referred to as the return loss. Kern proposed this equation to account for this pressure drop:[48]

$$\Delta P_r = \frac{4.0 \, n \, V^2}{S \, 2g}$$

in Appendix A nomenclature:

$$PLTR = \frac{4.0 \, NP \, VT^2}{\left(\frac{DENT}{DENW} \right)^2 \, AG} \cdot .4335$$

where .4335 is a conversion factor from feet of water to pounds per square inch and the ratio $\left(\frac{DENT}{DENW} \right)$ is the specific gravity of the fluid. Manufacturers of heat exchangers use a modification of this return loss equation. The factor of 4.0 was determined to provide results that were much too conservative. Practice is to use a factor of 2.0 in calculations.[49] The equation used in PRESCK does in fact use this factor of 2.0.

The total tube-side pressure drop is therefore:

$$PLTOT = PLTF + PLTR$$

This total tube-side pressure loss is compared to the maximum allowable tube-side pressure loss imposed by the user. If the tube-side pressure loss is greater than the allowable loss, the exchanger configuration is not deemed acceptable.

The last check is associated with the shell-side pressure loss. The pressure drop in a shell and tube heat exchanger is proportional to the number of times the fluid crosses the tube bundle between tube sheets. It is also proportional to the distance the fluid flows across the tube bundle. The calculation of the pressure loss can be made from:[50]

$$\Delta P_s = \frac{f G^2 D_s (N+1)}{2 g \rho D_e \left(\frac{\mu}{\mu_w} \right)^{0.14}}$$

where N is the number of baffles in the heat exchanger, D_e is the equivalent diameter of the shell cross flow channel, and f is the shell-side friction factor. This friction factor is calculated by a subroutine called SFRICT. A listing of this subroutine is in Appendix F. Subroutine SFRICT contains data on shell-side friction factors for bundles with 25% cut segmental baffles.[51] Like TFRICT, this subroutine uses a linear interpolation algorithm to calculate the fluid friction factor depending on the Reynolds Number of the shell-side fluid.

The shell-side pressure loss equation using nomenclature from Appendix A takes the form:

$$PLSTOT = \frac{FRS \cdot L \cdot GS^2 \left(\frac{L \times 12.0}{DSI \times PICKS} + 1 \right) \cdot .006944}{2 \cdot AG \cdot DENS \cdot DEQS \left(\frac{DVS}{DVFS} \right)^{0.14}}$$

where PICKS is the choice of baffle spacing, FRS is the shell-side fluid friction factor calculated in subroutine SFRICT, and .006944 is a conversion factor from pounds per square feet to pounds per square inch. This total shell-side pressure loss is compared to the maximum allowable shell-side pressure loss imposed by the user. If the calculated loss is less than the maximum allowable pressure loss the heat exchanger configuration is regarded as acceptable.

Finally, if the exchanger configuration successfully meets the length restriction, the tube-side pressure loss restriction and the shell-side pressure loss restriction it is considered as an acceptable exchanger configuration that will meet the limitations of the problem input by the user.

0. Output of Possible Exchanger Configurations

All heat exchanger configurations which meet the required heat duty of the problem and are within the length and pressure drop restrictions are output to the user. The output is performed by calling four subroutines, PRINT2, PRINT4, PRINT6, and PRINT8. Each of these subroutines correspond to their particular pass-construction type. A listing of these subroutines are in Appendix G.

If no exchanger configuration of a particular pass-construction meets the problem specifications, a verbal output to that effect is printed.

If no exchanger configuration of any pass-construction category meets the problem specifications, a message is printed which states that the length and pressure restriction cannot be met and the user must use split shell design or multiple shells in series. This computer program is not capable of handling those types of design.

P. User Selection of Possible Exchanger Configurations

Once the program has listed all the possible heat exchanger configurations by pass-construction, the user is asked to make a selection of a heat exchanger configuration. The user is reminded that only the listed configurations will meet the problem specifications.

The user responds to three input requests which pertain to his choice of exchanger configurations. The need only respond to the questions posed by the program. The first input is the shell diameter. The second input is the number of tube-side passes. The final input is the baffle spacing. Once the user specifies the choice of heat exchanger configurations, the program calculates the final output data.

Q. Calculation of Estimated Capital Cost

Many factors enter into how much a shell and tube heat exchanger may cost. The cost of engineering and design, raw materials, and labor are only three of the many parameters that must be explored. Each shell and tube heat exchanger manufacturer has its own method for doing cost calculations for a specific exchanger design.

One would assume that the length of a heat exchanger, as well as the shell diameter would affect the cost. Since material costs play an important role, the number of tubes would also seem to have an effect on

THE AVAILABLE HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

THE AVAILABLE 2-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 14.00000 INCHES:
WITH A 1.000000 BAFFLE SPACING
SHELL DIAMETER 18.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 20.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING

THE AVAILABLE 4-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 18.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 20.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 22.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 24.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 26.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 28.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING

THE AVAILABLE 6-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 22.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 24.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 26.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 28.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 30.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 32.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING
SHELL DIAMETER 34.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING

THE AVAILABLE 8-PASS HEAT EXCHANGER
CONFIGURATIONS ARE AS FOLLOWS:

| | |
|------|---------------------------|
| 9.0 | CHROME MOLY STEEL (2.25%) |
| 10.0 | CHROME MOLY STEEL (5%) |
| 11.0 | CHROME MOLY STEEL (12%) |
| 12.0 | STAINLESS STEEL (18%) |
| 13.0 | STAINLESS STEEL (25%) |
| 14.0 | ADMIRALTY METAL |
| 15.0 | COPPER |
| 16.0 | NICKEL |
| 17.0 | ZIRCONIUM |
| 18.0 | TITANIUM |

PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF MATERIAL DESIRED.(ie 1.0 6.0 18.0 etc.)

6.0

DO YOU WANT TO CHANGE THE TUBE FLUID CHOICE ?

N

DO YOU WANT TO CHANGE THE SHELL FLUID CHOICE ?

N

DO YOU WANT TO CHANGE THE TUBE FLUID TEMPS ?

N

DO YOU WANT TO CHANGE THE TUBE FLUID FLOW RATE ?

N

DO YOU WANT TO CHANGE THE SHELL FLUID TEMP ?

N

DO YOU WANT TO CHANGE THE SHELL FLUID FLOW RATE ?

N

DO YOU WANT TO CHANGE THE DECISION ON WHETHER OR NOT A FOULING FACTOR IS USED ?

N

DO YOU WANT TO CHANGE THE LENGTH RESTRICTION ?

N

DO YOU WANT TO CHANGE THE MAXIMUM ALLOWABLE PRESSURE LOSSES ?

N

NUMBER OF TUBE-SIDE PASSES: 8.0

BAFFLE SPACING: 0.20 OR

6.80 INCHES

EXCHANGER TUBE LENGTH: 1.461 FEET

TUBE-SIDE PRESSURE LOSS: 1.34 POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS: 0.18 POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST: \$ 418562.90

U = 194.6903

UCALC = 243.1180

DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

N

***** SINCE COPPER TUBING IS RELATIVELY EXPENSIVE, THE
***** TUBING MATERIAL WILL BE CHANGED TO REGULAR CARBON STEEL
***** TO SEE THE EFFECT ON THE FINAL RESULTS. ALL OTHER
***** PARAMETERS WILL BE LEFT UNCHANGED FROM THE ORIGINAL
***** PROBLEM STATEMENT.

DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS
TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO

Y

DO YOU WANT TO CHANGE THE TUBING SIZE ?

N

DO YOU WANT TO CHANGE THE TUBING MATERIAL ?

Y

WHAT MATERIAL SPECIFICATIONS DO YOU WANT FOR
THE TUBING ?

| CHOICE NUMBER | MATERIAL TYPE |
|---------------|------------------------|
| 1.0 | ALUMINUM TYPE 1100 |
| 2.0 | ALUMINUM TYPE 3003 |
| 3.0 | ALUMINUM TYPE 3004 |
| 4.0 | ALUMINUM TYPE 6061 |
| 5.0 | ALUMINUM TYPE 6063 |
| 6.0 | CARBON STEEL |
| 7.0 | CARBON MOLY STEEL |
| 8.0 | CHROME MOLY STEEL (1%) |

2.0, 4.0, 6.0, OR 8.0

2.0

WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
0.2, 0.45, OR 1.0

1.0

THE FINAL CONFIGURATION IS :

SHELL DIAMETER: 14.0 INCHES

NUMBER OF TUBE-SIDE PASSES: 2.0

BAFFLE SPACING: 1.00 OR

14.00 INCHES

EXCHANGER TUBE LENGTH: 8.205 FEET

TUBE-SIDE PRESSURE LOSS: 1.59 POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS: 3.54 POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST: \$ 46197.14

U = 194.6903
UCALC = 271.4765

DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

Y

WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
14.0, 28.0 etc.

***** THE LARGEST CONFIGURATION WILL BE EXAMINED HERE.

34.0

WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
2.0, 4.0, 6.0, OR 8.0

8.0

WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
0.2, 0.45, OR 1.0

0.2

THE FINAL CONFIGURATION IS :

SHELL DIAMETER: 34.0 INCHES

SHELL DIAMETER 26.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 28.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING

THE AVAILABLE 6-PASS HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 22.00000 INCHES:
 WITH A 0.4500000 BAFFLE SPACING
 SHELL DIAMETER 24.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 26.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 28.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 30.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 32.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 34.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING

THE AVAILABLE 8-PASS HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 26.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 28.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 30.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 32.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 34.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING

THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE
 OF THE ABOVE LISTED HEAT EXCHANGER
 CONFIGURATIONS. REMEMBER ONLY THESE LISTED
 CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU
 HAVE INPUT INTO THE PROGRAM

WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
 14.0, 26.0 etc.

***** THE SMALLEST CONFIGURATION WILL BE EXAMINED HERE.

14.0

WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE

TRANSFER COEFFICIENT GIVEN FOR CLEAN TUBES ?
ANSWER Y FOR YES AND N FOR NO.

Y

WHAT IS THE MAXIMUM LENGTH THAT THE HEAT EXCHANGER TUBES CAN BE FOR THIS APPLICATION ? REMEMBER, THE SHELL WILL HAVE TO BE REMOVED FOR CLEANING SO THE MAXIMUM LENGTH OF THE HEAT EXCHANGER TUBES SHOULD BE LESS THAN ONE HALF THE LENGTH OF THE ROOM THAT THE HEAT EXCHANGER WILL BE OPERATING IN. TYPE IN THE MAXIMUM LENGTH IN FEET THAT THE HEAT EXCHANGER TUBES CAN BE.
(ie 5.0 , 7.5, 19.6 etc.)

10.0

WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN THE TUBES IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)

5.0

WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN THE SHELL IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)

5.0

THE AVAILABLE HEAT EXCHANGER CONFIGURATIONS ARE AS FOLLOWS:

THE AVAILABLE 2-PASS HEAT EXCHANGER CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 14.00000 INCHES:
WITH A 1.000000 BAFFLE SPACING
SHELL DIAMETER 18.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 20.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING

THE AVAILABLE 4-PASS HEAT EXCHANGER CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 18.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 20.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 22.00000 INCHES:
WITH A 0.450000 BAFFLE SPACING
SHELL DIAMETER 24.00000 INCHES:
WITH A 0.200000 BAFFLE SPACING

| | |
|-----|------------------------|
| 5.0 | KEROSENE |
| 6.0 | SAE 10 LUBRICATING OIL |
| 7.0 | ETHYLENE GLYCOL |

PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED
TUBE FLUID. (ie 1.0 IF THE FLUID IS WATER)

1.0

WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN
THE SHELL OF THE EXCHANGER? THE FOLLOWING
FLUID CHOICES ARE AVAILABLE:

| CHOICE NUMBER | FLUID NAME |
|---------------|------------------------|
| 1.0 | WATER |
| 2.0 | METHYL ALCOHOL |
| 3.0 | GASOLINE |
| 4.0 | DOWTHERM A |
| 5.0 | KEROSENE |
| 6.0 | SAE 10 LUBRICATING OIL |
| 7.0 | ETHYLENE GLYCOL |

PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED
SHELL FLUID. (ie 1.0 IF THE FLUID IS WATER)

1.0

WHAT IS THE ENTERING TEMPERATURE OF THE TUBE
FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)

160.0

WHAT IS THE EXIT TEMPERATURE OF THE TUBE FLUID
IN DEGREES F ?

180.0

WHAT IS THE FLOW RATE OF THE TUBE FLUID IN
GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)

400.0

WHAT IS THE ENTERING TEMPERATURE OF THE SHELL
FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)

240.0

WHAT IS THE FLOW RATE OF THE SHELL FLUID IN
GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)

300.0

DO YOU WANT TO ADD A FOULING FACTOR TO THE HEAT

| | | |
|------|------|----|
| 29.0 | 1.25 | 16 |
| 30.0 | 1.25 | 17 |
| 31.0 | 1.25 | 18 |
| 32.0 | 1.25 | 20 |
| 33.0 | 1.5 | 10 |
| 34.0 | 1.5 | 12 |
| 35.0 | 1.5 | 14 |
| 36.0 | 1.5 | 16 |

PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF TUBING DESIRED.(ie 1.0 6.0 34.0 etc.)

9.0

WHAT MATERIAL SPECIFICATIONS DO YOU WANT FOR THE TUBING ?

| CHOICE NUMBER | MATERIAL TYPE |
|---------------|---------------------------|
| 1.0 | ALUMINUM TYPE 1100 |
| 2.0 | ALUMINUM TYPE 3003 |
| 3.0 | ALUMINUM TYPE 3004 |
| 4.0 | ALUMINUM TYPE 6061 |
| 5.0 | ALUMINUM TYPE 6063 |
| 6.0 | CARBON STEEL |
| 7.0 | CARBON MOLY STEEL |
| 8.0 | CHROME MOLY STEEL (1%) |
| 9.0 | CHROME MOLY STEEL (2.25%) |
| 10.0 | CHROME MOLY STEEL (5%) |
| 11.0 | CHROME MOLY STEEL (12%) |
| 12.0 | STAINLESS STEEL (18%) |
| 13.0 | STAINLESS STEEL (25%) |
| 14.0 | ADMIRALTY METAL |
| 15.0 | COPPER |
| 16.0 | NICKEL |
| 17.0 | ZIRCONIUM |
| 18.0 | TITANIUM |

PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF MATERIAL DESIRED.(ie 1.0 6.0 18.0 etc.)

15.0

WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN THE TUBES OF THE EXCHANGER? THE FOLLOWING FLUID CHOICES ARE AVAILABLE:

| CHOICE NUMBER | FLUID NAME |
|---------------|----------------|
| 1.0 | WATER |
| 2.0 | METHYL ALCOHOL |
| 3.0 | GASOLINE |
| 4.0 | DOWTHERM A |

WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.

Y

THE MAJOR SOURCES OF INFORMATION USED TO FORMULATE THIS PROGRAM ARE:

BREESE, J. - SHELL AND TUBE HEAT EXCHANGERS
 KAYS & LONDON - COMPACT HEAT EXCHANGERS
 KERN, D. - PROCESS HEAT TRANSFER
 HOLMAN, J.P. - HEAT TRANSFER
 CHASE, T.A. - PERSONAL COMMUNICATION
 FANARITIS, J.P. - HEAT EXCHANGER TECHNOLOGY
 GAGGIOLI, R.A. - EFFICIENCY AND COSTING
 PATTERSON-KELLEY CO. - HEAT EXCHANGERS
 TEMA - STANDARDS OF TUBULAR EXCHANGERS

WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.

Y

WHAT SIZE OF TUBING DO YOU WANT TO USE IN YOUR HEAT EXCHANGER? (REMEMBER THE SMALLER THE BWG NUMBER THE THICKER THE WALL OF THE TUBING)

| CHOICE NUMBER | OUTSIDE DIAMETER(IN) | BWG # |
|---------------|----------------------|-------|
| 1.0 | 0.75 | 10 |
| 2.0 | 0.75 | 11 |
| 3.0 | 0.75 | 12 |
| 4.0 | 0.75 | 13 |
| 5.0 | 0.75 | 14 |
| 6.0 | 0.75 | 15 |
| 7.0 | 0.75 | 16 |
| 8.0 | 0.75 | 17 |
| 9.0 | 0.75 | 18 |
| 10.0 | 0.75 | 19 |
| 11.0 | 0.75 | 20 |
| 12.0 | 1.0 | 8 |
| 13.0 | 1.0 | 10 |
| 14.0 | 1.0 | 11 |
| 15.0 | 1.0 | 12 |
| 16.0 | 1.0 | 13 |
| 17.0 | 1.0 | 14 |
| 18.0 | 1.0 | 15 |
| 19.0 | 1.0 | 16 |
| 20.0 | 1.0 | 18 |
| 21.0 | 1.0 | 20 |
| 22.0 | 1.25 | 7 |
| 23.0 | 1.25 | 8 |
| 24.0 | 1.25 | 10 |
| 25.0 | 1.25 | 11 |
| 26.0 | 1.25 | 12 |
| 27.0 | 1.25 | 13 |
| 28.0 | 1.25 | 14 |

IX. COMPUTER RESULTS

***** THIS IS A COPY OF THE ACTUAL RUN OF THE EXAMPLE
 ***** PROBLEM. THE ONLY ADDITIONS MADE ARE THE
 ***** COMMENTS INSERTED FOR CLARITY WHICH ARE PRECEDED
 ***** WITH FIVE STARS.

\$ SHELL

THIS PROGRAM CALCULATES THE REQUIRED LENGTH, SIZE OF SHELL, BAFFLE SPACING, NUMBER OF TUBE PASSES, AND ESTIMATED CAPITAL COST OF A TEMA TYPE E SHELL AND TUBE HEAT EXCHANGER WITH 2, 4, 6, OR 8 TUBE PASSES. THIS PROGRAM ALSO PROVIDES CALCULATIONS OF THE SHELL-SIDE AND TUBE-SIDE PRESSURE DROPS. THE USER PICKS THE SIZE OF THE TUBING TO BE USED ALONG WITH THE TUBE MATERIAL. THE USER SETS THE MAXIMUM LENGTH ALONG WITH THE MAXIMUM ALLOWABLE PRESSURE DROPS ON THE SHELL AND TUBE SIDES.

THE PROGRAM OUTPUTS THE EXCHANGER CONFIGURATIONS THAT WILL MEET THE PROBLEM LIMITATIONS. FINAL DESIGN DATA ON ANY CONFIGURATION IS PROVIDED AS REQUESTED. ALTERNATE EXCHANGER CONFIGURATIONS CAN BE COMPARED, AND ORIGINAL INPUT DATA CAN BE CHANGED TO DETERMINE THEIR EFFECTS ON THE FINAL DESIGN DATA.

THIS PROGRAM IS LIMITED TO LIQUID-TO-LIQUID APPLICATIONS UNDER LOW PRESSURE CONDITIONS (<300 PSI).

A LIMITED NUMBER OF COMMON FLUIDS ARE AVAILABLE TO BE CHOSEN AS FLUIDS IN THE TUBE AND SHELL. THIS PROGRAM CAN BE EASILY MODIFIED TO ACCEPT ADDITIONAL FLUIDS FOR WHICH DENSITY, VISCOSITY, THERMAL CONDUCTIVITY AND SPECIFIC HEAT DATA ARE AVAILABLE.

THE USER MUST INPUT THE FOLLOWING DATA IN THE UNITS SPECIFIED:

| INPUT DATA | UNITS |
|---------------------------|------------------------|
| TUBE FLUID ENTRANCE TEMP | DEGREES F |
| TUBE FLUID EXIT TEMP | DEGREES F |
| TUBE FLUID FLOW RATE | GALLONS PER MINUTE |
| SHELL FLUID ENTRANCE TEMP | DEGREES F |
| SHELL FLUID FLOW RATE | GALLONS PER MINUTE |
| MAXIMUM TUBE LENGTH | FEET |
| MAX TUBE PRESSURE LOSS | POUNDS PER SQUARE INCH |
| MAX SHELL PRESSURE LOSS | POUNDS PER SQUARE INCH |

VIII. SAMPLE PROBLEM

In order to picture just what this program actually accomplishes, a sample problem will be examined. The example comes from an article listed as Reference 30.

Problem Conditions:

| | |
|--|-----------------------------------|
| Tube fluid: water | Shell fluid: water |
| Tube entrance temperature: 160°F | Shell entrance temperature: 240°F |
| Tube exit temperature: 180°F | |
| Tube fluid flow rate: 400 GPM | Shell fluid flow rate: 300 GPM |
| Fouling factor will be added | |
| Maximum allowable tube length: 10 feet | |
| Tube outside diameter: 0.75 in | |
| Tube pitch: 15/16 in | |
| Tube layout design: equilateral triangle | |
| Tube thickness: 18 BWG | |
| Tube material: copper | |

No pressure loss limitations were mentioned in this example. In order to make this example meet the program's required inputs the following additional limitations will be imposed:

Maximum allowable tube-side pressure loss: 5.0 psi
Maximum allowable shell-side pressure loss: 5.0 psi

The actual inputs to the computer program and the results obtained are found in Chapter IX.

Baffle spacing
Exchanger tube length
Tube-side pressure loss
Shell-side pressure loss
Estimated capital cost

Each parameter is printed with its corresponding units. The user may request final design data on any or all of the listed exchanger configurations that meet the problem specifications.

S. User Directed Changes to Input Parameters

The user has the option of changing any or all of the input parameters. The program will query whether the user wants to change any of the initial inputs to the program. If the user responds affirmatively, the program responds with eleven questions representing all the inputs to the program. For example, the first question is, "Do you want to change the tubing size?" if the user responds "yes," the available tubing sizes will be reprinted on the screen and the user will be asked to make a choice of the size of tubing desired. The user may change all inputs or only selective inputs to the program. The parameters that are not changed will retain the value set by the initial problem input.

This aspect of the program is useful in that it enables the user to see how changes of tube size, fluid choices or temperature effect the list of possible heat exchanger configurations and the final design data.

Only the last of the preceeding conditions would impose a constraint on the flexibility of the program. There are eighteen choices of materials available to the user of the program, only one of which is 316 stainless steel. The solution to this problem was to assign cost factors to each of the available materials. Stainless steel (25%) was given a cost factor of 1.0 corresponding to 316 stainless steel. All other materials were given a cost factor higher or lower depending on cost of that particular material taken from a mechanical design text.[53]

Since this computer-aided design program only uses single shell pass construction with no additional shells in series or parallel, the cost equation, in Appendix A terminology reduces to:

For 2-pass construction:

$$Z = CF \cdot 1150 \left(\frac{DSI}{12.0} \right)^{1.05} L^{0.3} \left(\frac{TPH}{12.0} \right)^{0.75} NT_2^{0.975}$$

where CF is the cost factor mentioned previously. The equations for 4-, 6-, or 8-pass constructions are similar.

It should be noted that this cost estimate is crude at best. It does however, give the user another quantitative factor from which his choice of optimum heat exchanger configurations can be made.

R. Final Design Output

The user is provided the following data for each heat exchanger configuration he chooses to meet the problem specifications:

Shell diameter

Number of the tube-side passes

the overall cost. One equation which gives an estimate of the capital cost of a shell and tube heat exchanger in terms of several geometric variables is:[52]

$$Z = 1150 D^{1.05} L^{0.3} P^{0.75} N N_i^{0.975} N_e^{0.1} N_s N_p$$

where

Z is cost in dollars

D is outside tube diameter (ft)

L is tube length (ft)

P is tube pitch (ft)

N is number of tubes per pass

N_i is number of tube passes

N_e is number of shell passes

N_s is number of shells in series

N_p is number of shells in parallel

The use of this equation is limited to the following conditions:

- Length between three and thirty feet
- Tube diameters between 3/8 and 1.5 inches
- Tube pitch between 1.25 and 1.5 of the tube diameter
- Heat transfer area between 250 and 5000 square feet
- Fixed tube sheet construction
- Operating pressures less than 300 psi
- Material 316 stainless steel

SHELL DIAMETER 26.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 28.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 30.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 32.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 34.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING

***** NOTE THAT THE LIST OF AVAILABLE CONFIGURATIONS OF
 ***** HEAT EXCHANGERS IS NOT EFFECTED BY THE CHANGE IN
 ***** MATERIAL SELECTION. THE ESTIMATED COSTS WILL
 ***** BE SIGNIFICANTLY CHANGED HOWEVER.

THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE
 OF THE ABOVE LISTED HEAT EXCHANGER
 CONFIGURATIONS. REMEMBER ONLY THESE LISTED
 CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU
 HAVE INPUT INTO THE PROGRAM

WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
 14.0, 28.0 etc.

***** THE SMALLEST CONFIGURATION WILL BE EXAMINED HERE.

14.0

WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
 2.0, 4.0, 6.0, OR 8.0

2.0

WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
 0.2, 0.45, OR 1.0

1.0

THE FINAL CONFIGURATION IS :

SHELL DIAMETER: 14.0 INCHES

NUMBER OF TUBE-SIDE PASSES: 2.0

BAFFLE SPACING: 1.00 OR

14.00 INCHES

EXCHANGER TUBE LENGTH: 8.814 FEET

TUBE-SIDE PRE3SURE LOSS: 1.66 POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS: 3.80 POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST: \$ 4285.43

U = 194.6903
UCALC = 252.7155

DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

Y

WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
14.0, 28.0 etc.

***** THE LARGEST CONFIGURATION WILL BE EXAMINED HERE.

34.0

WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
2.0, 4.0, 6.0, OR 8.0

8.0

WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
0.2, 0.45, OR 1.0

0.2

THE FINAL CONFIGURATION IS :

SHELL DIAMETER: 34.0 INCHES

NUMBER OF TUBE-SIDE PASSES: 8.0

BAFFLE SPACING: 0.20 OR

6.80 INCHES

EXCHANGER TUBE LENGTH: 1.558 FEET

TUBE-SIDE PRESSURE LOSS: 1.36 POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS: 0.20 POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST: \$ 38743.28

U = 194.6903
UCALC = 227.9624

DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

N

***** A DEMONSTRATION OF HOW A CHANGE IN THE LENGTH
***** RESTRICTION EFFECTS THE FINAL RESULTS WILL BE
***** MADE BY CHANGING THE MAXIMUM LENGTH TO 2.0 FEET.

DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS
TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO

Y

DO YOU WANT TO CHANGE THE TUBING SIZE ?

N

DO YOU WANT TO CHANGE THE TUBING MATERIAL ?

N

DO YOU WANT TO CHANGE THE TUBE FLUID CHOICE ?

N

DO YOU WANT TO CHANGE THE SHELL FLUID CHOICE ?

N

DO YOU WANT TO CHANGE THE TUBE FLUID TEMPS ?

N

DO YOU WANT TO CHANGE THE TUBE FLUID FLOW RATE ?

N

DO YOU WANT TO CHANGE THE SHELL FLUID TEMP ?

N

DO YOU WANT TO CHANGE THE SHELL FLUID FLOW RATE ?

N

DO YOU WANT TO CHANGE THE DECISION ON WHETHER OR
NOT A FOULING FACTOR IS USED ?

N

DO YOU WANT TO CHANGE THE LENGTH RESTRICTION ?

Y

DO YOU WANT TO CHANGE THE MAXIMUM ALLOWABLE
PRESSURE LOSSES ?

N

WHAT IS THE MAXIMUM LENGTH THAT THE HEAT EXCHANGER TUBES CAN BE FOR THIS APPLICATION ?
 REMEMBER, THE SHELL WILL HAVE TO BE REMOVED FOR CLEANING SO THE MAXIMUM LENGTH OF THE HEAT EXCHANGER TUBES SHOULD BE LESS THAN ONE HALF THE LENGTH OF THE ROOM THAT THE HEAT EXCHANGER WILL BE OPERATING IN. TYPE IN THE MAXIMUM LENGTH IN FEET THAT THE HEAT EXCHANGER TUBES CAN BE.
 (ie 5.0 , 7.5, 19.6 etc.)

2.0

NO 2-PASS HEAT EXCHANGER CONFIGURATIONS
 WILL MEET THE PROBLEM SPECIFICATIONS.

NO 4-PASS HEAT EXCHANGER CONFIGURATIONS
 WILL MEET THE PROBLEM SPECIFICATIONS.

THE AVAILABLE HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

THE AVAILABLE 6-PASS HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 30.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 32.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 34.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING

THE AVAILABLE 8-PASS HEAT EXCHANGER
 CONFIGURATIONS ARE AS FOLLOWS:

SHELL DIAMETER 30.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 32.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING
 SHELL DIAMETER 34.00000 INCHES:
 WITH A 0.2000000 BAFFLE SPACING

***** NOTE THAT THE LIST OF AVAILABLE EXCHANGER
 CONFIGURATIONS IS GREATLY REDUCED.

THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE
 OF THE ABOVE LISTED HEAT EXCHANGER
 CONFIGURATIONS. REMEMBER ONLY THESE LISTED
 CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU

HAVE INPUT INTO THE PROGRAM

WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,
14.0, 28.0 etc.

30.0

WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE
2.0, 4.0, 6.0, OR 8.0

6.0

WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE
0.2, 0.45, OR 1.0

0.2

THE FINAL CONFIGURATION IS :

SHELL DIAMETER: 30.0 INCHES

NUMBER OF TUBE-SIDE PASSES: 6.0

BAFFLE SPACING: 0.20 OR

6.00 INCHES

EXCHANGER TUBE LENGTH: 1.929 FEET

TUBE-SIDE PRESSURE LOSS: 1.01 POUNDS PER SQUARE INCH

SHELL-SIDE PRESSURE LOSS: 0.48 POUNDS PER SQUARE INCH

ESTIMATED CAPITAL COST: \$ 28282.48

U = 194.6903

UCALC = 237.3057

DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER
CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.

N

DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS
TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO

N

DO YOU WANT TO DO ANY OTHER HEAT EXCHANGER
PROBLEMS ? TYPE Y FOR YES AND N FOR NO.

N

Fortran STOP DONE

X. CONCLUSION

A. Discussion

The major intent of this thesis was to create a computer program which would provide the user with quick preliminary sizes of shell and tube heat exchangers which will accomplish the required heat duty while meeting the length and pressure drop limitations imposed. This program was developed with information and equations that are publicly available. No proprietary code was used. No attempt has been made to compete with the more sophisticated software of proprietary organizations such as HTRI and Heat Transfer and Fluid Flow Services (HTFS). It is the shell and tube exchanger manufacturer's job to squeeze every last BTU out of a given heat exchanger design and to use every last tenth of a PSI to shorten the length of an exchanger. Their reason is obvious; to give the buyer the cheapest, and most effective heat exchanger they can. While the program that has been developed will not provide the user the information needed to completely build a shell and tube heat exchanger, it does provide with reasonable accuracy the key information that one needs to know: the exchanger length, the exchanger shell diameter, the shell-side pressure loss, the tube-side pressure loss and the estimated capital cost. Armed with this information, one can seek out a manufacturer that will build the heat exchanger to specifications.

The program developed is very user-friendly. The user can continually change input parameters to see how the changes affect the final design options provided. This aspect gives the program some educational significance. The user can see what happens when a more viscous fluid is exchanged for a less viscous one. The addition or deletion of the

fouling factor from the design input can also give the user some insight into just what a scaling deposit does to the heat transfer in an exchanger.

The program does indeed accomplish what it was expected to do. The time-intensive and laborious calculations needed to design a single heat exchanger have been organized into a compact program which provides nearly instantaneous results to the user.

B. Suggestions for Improvement

This program is suited to further expansion and revision. The program is limited to fixed tube sheet type "E" shells, with triangular tube pitch layout under single-phase liquid-to-liquid low pressure operations. Any reductions in these limiting features would greatly enhance the program capabilities. There is sufficient experimental data, equations, and commercial usage data available to widen the scope of the programs uses. The program could be expanded to handle TEMA type "F", "G", "H", "J", or "X" shells. Square and rotated square tube pitch layouts could be incorporated. Multiple-phase design can be added as well as gas-to-gas and gas-to-liquid operations. High pressure operations could be included.

The program can be expanded to also do more of the mechanical design. All of the integral parts of the shell and tube heat exchanger could be designed: the thickness of the tubesheets, the thickness of the shell, the size and type of gaskets, the size of the end covers and the number, type and size of bolts needed to name just a few.

Other improvements may be to increase the selection of fluids available and to allow for the evaluation of parallel and series heat exchanger configurations.

REFERENCES

1. Kakac, S., Bergeles, A.E., and Mayinger, F. Eds Heat Exchangers: Thermal-Hydraulic Fundamentals and Design. New York, McGraw-Hill Book Company, (1981), p. 537.
2. TEMA, Standards of Tubular Exchanger Manufacturers Association, 6th Ed. 1978, p. 135.
3. Singh, Krishna P. and Soler, Alan I., Mechanical Design of Heat Exchangers. New Jersey, Arcturus Publishers Inc., (1984), p. 16.
4. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
5. Chase, A. Timothy, "Mechanical Design of Heat Transfer Equipment" Presentation to Philadelphia and Delaware Sections ASME, March 19, 1981.
6. Kakac, S., p. 542.
7. Fanaritis, J.P., Kwas, J.A., and Chase, A.T., "Heat Exchange Technology" Encyclopedia of Chemical Technology, New York, John Wiley & Sons, Inc., (1980), Volume 12, Third Edition, p. 155.
8. Taborek, J., "Evolution of Heat Exchanger Design Techniques," Heat Transfer Engineering, (1979), Vol. 1., p. 15.
9. Fraas, A.P. and Ozisik, M.N., Heat Exchanger Design. New York, John Wiley & Sons, Inc., (1965), p. 151.
10. Kakac, S., pp. 581-588.

11. Tinker, T. "Shell-Side Characteristics of Shell and Tube Heat Exchangers," Trans ASME, Vol. 80, 1958, pp. 36-49.
12. Fraas, A.P., pp. 151-155.
13. Rohsenow, W.M. and Hartnett, J.P. Eds., Handbook of Heat Transfer. New York, McGraw-Hill Book Company, (1973), pp. 18-45 to 18-50.
14. Afgan, N.H., and Schlunder, E.U., Eds., Heat Exchangers: Design and Theory Sourcebook. New York, McGraw-Hill Book Company, (1974), pp. 155-168.
15. Sherwood, T.K. "A Review of the Development of Mass Transfer Theory," Chemical Engineering Education, (1974), p. 49.
16. Taborek, J., pp. 15-16.
17. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
18. Coulson, J.M. and Richardson, J.F., Chemical Engineering. New York, MacMillan Company, (1964), pp. 259-260.
19. Patterson-Kelley Company, Heat Exchangers. Manual No. 700-A, 1959, pp. e8 - e9.
20. Adams, A.J. and Rogers D.F., Computer-Aided Heat Transfer Analysis. New York, McGraw-Hill Book Company, (1973), pp. 368-369.
21. TEMA, p. 29.
22. Singh, K. P., pp. 735-735.

23. Kacac, S., pp. 1090-1091.
24. TEMA, pp. 138-139.
25. Kacac, S. pp. 1013-1047.
26. Ibid., pp. 1091-1093.
27. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
28. Knutson, J.L., Customer Service Manager, Heresite-Saekaphen, Inc., Private Communication, 15 January 1985.
29. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
30. Breeze, James L. "Shell and Tube Heat Exchangers" ASHRAE Journal, Vol. 26, No. 5 (May 1984), pp. 24-27.
31. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
32. Ibid.
33. Breeze, James L., pp. 24-27.
34. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
35. Kern, Donald Q., Process Heat Transfer. New York, McGraw-Hill Book Company (1950), p. 129.

36. Holman, J.P., Heat Transfer. Fifth Edition. New York, McGraw-Hill Book Company, (1981), p. 456.
37. Kays, W.M. and London, A.L., Compact Heat Exchangers. Third Edition. New York, McGraw-Hill Book Company, (1984), p. 16.
38. Ibid.
39. Holman, J.P., p. 457.
40. Kays, W.M., p. 23.
41. Kakac, S., pp. 570-573.
42. Kern, Donald Q., pp. 138-139.
43. Fanaritis, J.P., p. 157.
44. Kern, Donald Q., p. 103.
45. TEMA, p. 137.
46. Ibid.
47. Kern, Donald Q., p. 896.
48. Ibid., p. 148.
49. Chase, A. Timothy, Manager, Special Products Department, Struthers Wells Corporation, Private Communication, 29 January 1985.
50. Kern, Donald Q., p. 147.
51. Ibid., p. 839.

52. Gaggioli, Richard A., Ed. Efficiency and Costing. Washington, D.C., American Chemical Society, (1983), pp. 232-233.
53. Shigley, J.E., Mechanical Engineering Design. New York, McGraw-Hill Book Company, Inc., (1963), p. 220.

APPENDIX A

VARIABLE LISTING

This is an alphabetical display of the variables used in the main computer program. Their explanation and units are provided for quick reference and numerical analysis checks.

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|---------------------|
| ACALC | Answer designation to questions asked in the program. | - |
| ACORR | Required heat transfer area (calculated). | FT ² |
| AFFAC | Answer designation to change the fouling factor choice. | - |
| AFOUL | Answer designation to fouling factor question asked in the program. | - |
| AG | Gravitational acceleration constant = 32.1740. | FT/SEC ² |
| AGAIN | Answer designation to changing input parameters asked in the program. | - |
| AL2(I) | An array of tube bundle surface areas per foot of tube length for 2-pass construction. | FT ² /FT |
| AL4(I) | An array of tube bundle surface areas per foot of tube length for 4-pass construction. | FT ² /FT |

AD-A153 779

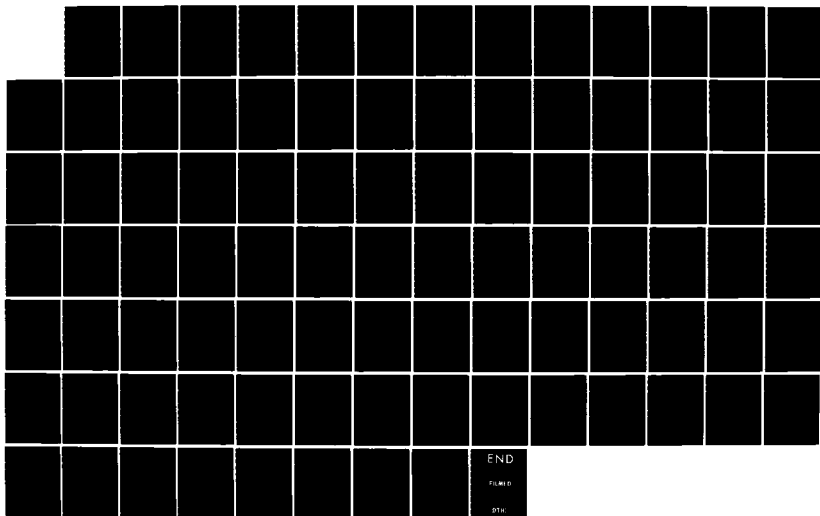
COMPUTER-AIDED THERMOHYDRAULIC DESIGN OF TEMA TYPE E
SHELL AND TUBE HEAT. (U) ARMY MILITARY PERSONNEL CENTER
ALEXANDRIA VA N J KOLAR 81 APR 85

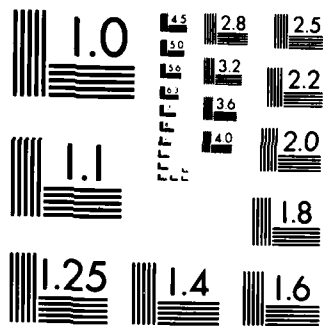
2/2

UNCLASSIFIED

F/G 13/1

NL





| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|---------------------|
| AL6(I) | An array of tube bundle surface areas per foot of tube length for 6-pass construction. | FT ² /FT |
| AL8(I) | An array of tube bundle surface areas per foot of tube length for 8-pass construction. | FT ² /FT |
| ALENTN | Answer designation to length restriction question asked in the program. | - |
| ALREST | Answer designation to change the length restriction. | - |
| AMATE | Answer designation to change the tube material. | - |
| AMAXSP | Maximum allowable shell-side pressure loss. | LB/IN ² |
| AMAXTP | Maximum allowable tube-side pressure loss. | LB/IN ² |
| APRES | Answer designation of change the allowable pressure drops. | - |
| ARAV | Maximum number of tubes possible depending on available area. | - |
| ASC | Crossflow area of shell | IN ² |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|---------------------|
| ASFLU | Answer designation to change the shell fluid. | - |
| ASGPM | Answer designation to change the shell-side flow rate. | - |
| ASTEM | Answer designation to change the shell-side temperature. | - |
| ATFLU | Answer designation to change the tube fluid. | - |
| ATGPM | Answer designation to change the tube-side flow rate. | - |
| ATI(I) | An array of available internal areas of tubes. | IN ² |
| ATIN | Choice of internal area of tube. | IN ² |
| ATS(I) | An array of available external surface areas of tubes per foot of tube length. | FT ² /FT |
| ATSU | Choice of external surface area of tube per foot of tube length. | FT ² /FT |
| ATTEM | Answer designation to change the tube-side temperatures. | - |
| ATUBE | Answer designation to change the tube size. | - |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|---|---------------------|
| C | Constant = .3208342 (conversion factor). | - |
| CASE | Number representing whether the tube fluid is being heated or cooled. | - |
| CF | Cost factor of the tube material choice. | - |
| CFM(I) | An array of cost factors depending on material type. | - |
| CMAX | Capacity rate of the maximum fluid. | BTU/HR-°F |
| CMIN | Capacity rate of the minimum fluid. | BTU/HR-°F |
| CP | Specific heat at constant pressure value calculated in subroutine. | BTU/LBM-°F |
| CPS | Specific heat at constant pressure of the shell fluid. | BTU/LBM-°F |
| CPT | Specific heat at constant pressure of the tube fluid. | BTU/LBM-°F |
| CS | Capacity rate of shell fluid. | BTU/HR-°F |
| CT | Capacity rate of tube fluid. | BTU/HR-°F |
| DEN | Density value calculated in subroutine | LBM/FT ³ |
| DENS | Density of shell fluid. | LBM/FT ³ |
| DENT | Density of tube fluid. | LBM/FT ³ |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|---------------------|
| DENW | Density of water for specific gravity calculations. | LBM/FT ³ |
| DEQS | Equivalent diameter of the fluid conduit crossing the tubes. | IN |
| DSI(I) | An array of available inside diameters of shells. | IN |
| DSIU(I) | An array of available usable diameters of shells. | IN |
| DTI(I) | An array of available inside diameters of tubes. | IN |
| DTIN | Choice of inside tube diameter. | IN |
| DTO(I) | An array of available outside diameters of tubes. | IN |
| DTOT | Choice of outside tube diameter. | IN |
| DV | Dynamic viscosity value calculated in subroutine. | LB/FT-SEC |
| DVS | Dynamic viscosity of shell fluid. | LB/FT-SEC |
| DVFS | Dynamic viscosity of shell fluid at at wall temperature. | LB/FT-SEC |
| DVT | Dynamic viscosity of tube fluid. | LB/FT-SEC |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|---|--------------|
| DVTF | Dynamic viscosity of tube fluid at wall temperature. | LB/FT-SEC |
| EFF | Heat transfer effectiveness. | - |
| F1 | Constant = 0.9069, which accounts for triangular tube arrangement in calculating the number of tubes possible for a given shell diameter. | - |
| F2 | Factor used in calculating the number of tubes possible for a given shell diameter for 2-pass construction. | - |
| F4 | Factor used in calculating the number of tubes possible for a given shell diameter for 4-pass construction. | - |
| F6 | Factor used in calculating the number of tubes possible for a given shell diameter for 6-pass construction. | - |
| F8 | Factor used in calculating the number of tubes possible for a given shell diameter for 8-pass construction. | - |
| FRS | Shell-side friction factor. | - |
| FRT | Tube-side friction factor. | - |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|---|
| GS | Mass velocity of the shell fluid. | LB/SEC-FT ² |
| GT | Mass velocity of the tube fluid. | LB/SEC-FT ² |
| HI | Heat transfer film coefficient inside the tubes. | BTU/HR-°F-FT ² |
| HO | Heat transfer film coefficient outside the tubes. | BTU/HR-°F-FT ² |
| I | Counter | - |
| J | Counter | - |
| J2 | Counter | - |
| J4 | Counter | - |
| J6 | Counter | - |
| J8 | Counter | - |
| K | Thermal conductivity value calculated in subroutine. | $\frac{\text{BTU}}{\text{HR-FT}^2\text{-°F}}$ |
| K1 | Thermal conductivity interim value used in subroutines. | $\frac{\text{BTU}}{\text{HR-FT}^2\text{-°F}}$ |
| K2 | Counter | - |
| K4 | Counter | - |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|--|
| K6 | Counter | - |
| K8 | Counter | - |
| KM(I) | An array of thermal conductivities of metals. | $\frac{\text{BTU-FT}}{\text{HR-FT}^2\text{-}^\circ\text{F}}$ |
| KS | Thermal conductivity of shell fluid. | - |
| KT | Thermal conductivity of tube fluid. | $\frac{\text{BTU}}{\text{HR-FT-}^\circ\text{F}}$ |
| KW | Choice of thermal conductivity of the tube metal. | $\frac{\text{BTU-FT}}{\text{HR-FT}^2\text{-}^\circ\text{F}}$ |
| L | Length of tubing of chosen heat exchanger configuration. | FT |
| LRESTR | Length restriction of the heat exchanger tubes. | FT |
| M1 | Counter | - |
| M2 | Counter | - |
| M3 | Counter | - |
| MS | Shell-side fluid flow rate. | GPM |
| MT | Tube-side fluid flow rate. | GPM |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|---|--------------|
| N | Integer designation of shell diameter choice. | - |
| NP | Number designation of number of tube passes. | - |
| NT2(I) | An array of tube counts for 2-pass construction. | - |
| NT4(I) | An array of tube counts for 4-pass construction. | - |
| NT6(I) | An array of tube counts for 6-pass construction. | - |
| NT8(I) | An array of tube counts for 8-pass construction. | - |
| NTU | Number of heat transfer units. | - |
| OKFIN | Counter representing final OK to print exchanger configuration. | - |
| PI | Constant = 3.1415927 | - |
| PICKD | Number designating the choice of shell diameter. | IN |
| PICKFS | Number designating the choice of fluid in the shell. | - |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|---|--------------------|
| PICKFT | Number designating the choice of fluid in the tubes. | - |
| PICKM | Number designating the choice of metal of the tubing. | - |
| PICKP | Number designating the choice of number of tube passes. | - |
| PICKS | Number designating the choice of baffle spacing. | - |
| PICKT | Number designating the choice of tubing size. | - |
| PLSTOT | Total shell-side pressure loss. | LB/IN ² |
| PLTF | Tube-side pressure loss due to friction losses. | LB/IN ² |
| PLTR | Tube-side pressure loss due to return losses. | LB/IN ² |
| PLTTOT | Total tube-side pressure loss. | LB/IN ² |
| Q | Heat transfer rate | BTU/HR |
| QSK1H(I) | An array of maximum shell-side flow rates for .2 baffle spacing construction. | GPM |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|--------------|
| QSK1L(I) | An array of minimum shell-side flow rates for .2 baffle spacing construction. | GPM |
| QSK2H(I) | An array of maximum shell-side flow rates for .45 baffle spacing construction. | GPM |
| QSK2L(I) | An array of minimum shell-side flow rates for .45 baffle spacing construction. | GPM |
| QSK3H(I) | An array of maximum shell-side flow rates for 1.0 baffle spacing construction. | GPM |
| QSK3L(I) | An array of minimum shell-side flow rates for 1.0 baffle spacing construction. | GPM |
| QTH2(I) | An array of maximum tube-side flow rates for 2-pass construction. | GPM |
| QTH4(I) | An array of maximum tube-side flow rates for 4-pass construction. | GPM |
| QTH6(I) | An array of maximum tube-side flow rates for 6-pass construction. | GPM |
| QTH8(I) | An array of maximum tube-side flow rates for 8-pass construction. | GPM |
| QTL2(I) | An array of minimum tube-side flow rates for a 2-pass construction. | GPM |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|---|---|
| QTL4(I) | An array of minimum tube-side flow rates for a 4-pass construction. | GPM |
| QTL6(I) | An array of minimum tube-side flow rates for a 6-pass construction. | GPM |
| QTL8(I) | An array of minimum tube-side flow rates for 8-pass construction. | GPM |
| RATIO | Ratio of the minimum capacity fluid to the maximum capacity fluid. | - |
| RES | Reynolds number of the shell fluid. | - |
| RET | Reynolds number of the tube fluid. | - |
| RFOULT | Total fouling resistance in the heat exchanger. | $\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$ |
| RI | Fouling resistance on inside of tubes. | $\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$ |
| RO | Fouling resistance on outside of tubes. | $\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$ |
| RW | Resistance of tube wall. | $\frac{\text{HR-FT}^2\text{-}^\circ\text{F}}{\text{BTU}}$ |
| SF(I) | An array of shell-side friction factors. | - |

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|---|--------------|
| SRE(I) | An array of shell-side fluid Reynolds numbers. | - |
| T1MAX | Entrance temperature of the maximum fluid. | °F |
| T1MIN | Entrance temperature of the minimum fluid. | °F |
| T2MAX | Exit temperature of the maximum fluid. | °F |
| T2MIN | Exit temperature of the minimum fluid. | °F |
| TAVE | Average temperature value used in subroutine calculations. | °F |
| TC | Tube clearance. | IN |
| TF(I) | An array of tube-side friction factors | - |
| TH | Thickness of the partition wall for multiple pass construction. | IN |
| P(I) | An array of available tube pitches of tubes. | IN |
| TPH | Choice of tube pitch. | IN |
| TRATIO | Ratio of outside to inside surface of tubing. | - |
| TRE(I) | An array of tube-side fluid Reynolds numbers. | - |


```

DO 16, I=1,28

  QSK1H(I) = 3.116875*.20*DSI(I)*DSIU(I)*(TC/TPH)*4.5

16  CONTINUE

***  CALCULATE THE LOW SHELL-SIDE FLOW RATE FOR A BAFFLE
***  SPACING OF .45*SHELL DIAMETER (K2) BASED ON A 2 FT/SEC
***  FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

DO 17, I=1,28

  QSK2L(I) = 3.116875*.45*DSI(I)*DSIU(I)*(TC/TPH)*2.0

17  CONTINUE

***  CALCULATE THE HIGH SHELL-SIDE FLOW RATE FOR A BAFFLE
***  SPACING OF .45*SHELL DIAMETER (K2) BASED ON A 4.5 FT/SEC
***  FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

DO 18, I=1,28

  QSK2H(I) = 3.116875*.45*DSI(I)*DSIU(I)*(TC/TPH)*4.5

18  CONTINUE

***  CALCULATE THE LOW SHELL-SIDE FLOW RATE FOR A BAFFLE
***  SPACING OF 1.0*SHELL DIAMETER (K3) BASED ON A 2 FT/SEC
***  FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

DO 19, I=1,28

  QSK3L(I) = 3.116875*1.0*DSI(I)*DSIU(I)*(TC/TPH)*2.0

19  CONTINUE

***  CALCULATE THE HIGH SHELL-SIDE FLOW RATE FOR A BAFFLE
***  SPACING OF 1.0*SHELL DIAMETER (K3) BASED ON A 4.5 FT/SEC
***  FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

DO 20, I=1,28

  QSK3H(I) = 3.116875*1.0*DSI(I)*DSIU(I)*(TC/TPH)*4.5

20  CONTINUE

***  CALCULATE THE CAPACITIES OF THE SHELL AND TUBE FLUIDS

***  DEPENDING ON THE TUBE FLUID CHOICE CALCULATE THE TUBE
***  FLUID PROPERTIES OF DENSITY, SPECIFIC HEAT, DYNAMIC
***  VISCOSITY AND THERMAL CONDUCTIVITY

TAVE = TTAVE

```

```

      QTL4(I) = 3.116875 * NT4(I) * ATIN * 2.0 / 4.0

11  CONTINUE

***  CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 4-PASS
***  EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
***  AVAILABLE SHELL DIAMETER

      DO 12, I=1,28

          QTH4(I) = 3.116875 * NT4(I) * ATIN * 8.0 / 4.0

12  CONTINUE

***  CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 2-PASS
***  EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
***  AVAILABLE SHELL DIAMETER

      DO 13, I=1,28

          QTL2(I) = 3.116875 * NT2(I) * ATIN * 2.0 / 2.0

13  CONTINUE

***  CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 2-PASS
***  EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
***  AVAILABLE SHELL DIAMETER

      DO 14, I=1,28

          QTH2(I) = 3.116875 * NT2(I) * ATIN * 8.0 / 2.0

14  CONTINUE

***  CALCULATE THE TUBE CLEARANCE - THE SPACING BETWEEN THE
***  TUBES - FROM THE DEFINITION OF TUBE PITCH

      TC = TPH - DTOT

***  CALCULATE THE LOW SHELL-SIDE FLOW RATE FOR A BAFFLE
***  SPACING OF .2*SHELL DIAMETER (K1) BASED ON A 2 FT/SEC
***  FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

      DO 15, I=1,28

          QSK1L(I) = 3.116875*.20*DSI(I)*DSIU(I)*(TC/TPH)*2.0

15  CONTINUE

***  CALCULATE THE HIGH SHELL-SIDE FLOW RATE FOR A BAFFLE
***  SPACING OF .2*SHELL DIAMETER (K1) BASED ON A 4.5 FT/SEC
***  FLUID VELOCITY FOR EACH AVAILABLE SHELL DIAMETER

```

DO 6, I=1,28

AL2(I) = NT2(I) * ATSU
AL4(I) = NT4(I) * ATSU
AL6(I) = NT6(I) * ATSU
AL8(I) = NT8(I) * ATSU

6 CONTINUE

*** CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 8-PASS
*** EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
*** AVAILABLE SHELL DIAMETER

DO 7, I=1,28

QTL8(I) = 3.116875 * NT8(I) * ATIN * 2.0 / 8.0

7 CONTINUE

*** CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 8-PASS
*** EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
*** AVAILABLE SHELL DIAMETER

DO 8, I=1,28

QTH8(I) = 3.116875 * NT8(I) * ATIN * 8.0 / 8.0

8 CONTINUE

*** CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 6-PASS
*** EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
*** AVAILABLE SHELL DIAMETER

DO 9, I=1,28

QTL6(I) = 3.116875 * NT6(I) * ATIN * 2.0 / 6.0

9 CONTINUE

*** CALCULATE THE HIGH TUBE-SIDE FLOW RATE FOR A 6-PASS
*** EXCHANGER BASED ON A 8 FT/SEC FLUID VELOCITY FOR EACH
*** AVAILABLE SHELL DIAMETER

DO 10, I=1,28

QTH6(I) = 3.116875 * NT6(I) * ATIN * 8.0 / 6.0

10 CONTINUE

*** CALCULATE THE LOW TUBE-SIDE FLOW RATE FOR A 4-PASS
*** EXCHANGER BASED ON A 2 FT/SEC FLUID VELOCITY FOR EACH
*** AVAILABLE SHELL DIAMETER

DO 11, I=1,28

IF (AGAIN .EQ. 'Y') THEN
GO TO 716
END IF

*** THE NUMBER OF TUBES ALLOWABLE IS CALCULATED FOR COMMON
*** SHELL DIAMETERS STARTING WITH 6 INCHES AND INCREASING
*** IN INCREMENTS OF 2 INCHES TO A MAXIMUM OF 60 INCHES

3 DO 5, I=1,28

IF(DSI(I) .GE. 6.0 .AND. DSI(I) .LE. 18.0) THEN
TH = 0.25
END IF

IF(DSI(I) .GT. 18.0 .AND. DSI(I) .LE. 36.0) THEN
TH = 0.50
END IF

IF(DSI(I) .GT. 36.0 .AND. DSI(I) .LE. 60.0) THEN
TH = 0.75
END IF

*** THE USABLE SHELL DIAMETER IS .5 INCHES LESS THAN THE
*** ACTUAL SHELL DIAMETER DUE TO TUBE SHEET CONSTRUCTION

DSIU(I) = (DSI(I) - 0.5)

F1 = 0.9069

F2 = 1.0 - ((TH * DSIU(I)) / ((PI/4.0) * (DSIU(I)**2.0)))

ARAV = ((DSIU(I) - DTOT) / TPH) ** 2.0

NT2(I) = ARAV * F1 * F2

F4 = 1.0 - ((2.125 * TH * DSIU(I)) / ((PI/4.0) * (DSIU(I)**2.0)))

NT4(I) = ARAV * F1 * F4

F6 = 1.0 - ((3.25 * TH * DSIU(I)) / ((PI/4.0) * (DSIU(I)**2.0)))

NT6(I) = ARAV * F1 * F6

F8 = 1.0 - ((4.0 * TH * DSIU(I)) / ((PI/4.0) * (DSIU(I)**2.0)))

NT8(I) = ARAV * F1 * F8

5 CONTINUE

*** CALCULATE THE AVAILABLE TUBE SURFACE AREA PER FOOT OF
*** LENGTH OF THE HEAT EXCHANGER FOR EACH AVAILABLE SHELL
*** DIAMETER

```

PRINT*, ' '
713 PRINT*, 'WHAT IS THE ENTERING TEMPERATURE OF THE SHELL'
PRINT*, 'FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)'
PRINT*, ' '

READ*, TS1

*** CASE 1 REPRESENTS THE TUBE FLUID BEING HEATED

IF (TT1 .LT. TT2) THEN
    CASE = 1.0
END IF

*** CASE 2 REPRESENTS THE TUBE FLUID BEING COOLED

IF (TT1 .GT. TT2) THEN
    CASE = 2.0
END IF

*** CHECK FOR IMPOSSIBLE TEMPERATURE CONDITIONS

IF (CASE .EQ. 1.0 .AND. TS1 .LE. TT2) THEN
    PRINT*, ' '
    PRINT*, 'IT IS IMPOSSIBLE TO HAVE THE TUBE FLUID BEING'
    PRINT*, 'HEATED AND THE SHELL FLUID ENTRANCE TEMPERATURE'
    PRINT*, 'LESS THAN THE TUBE FLUID EXIT TEMPERATURE. TRY'
    PRINT*, 'INPUTTING THE SHELL FLUID ENTRANCE TEMPERATURE'
    PRINT*, 'AGAIN.'
    PRINT*, ' '
    GO TO 713
END IF

IF (CASE .EQ. 2.0 .AND. TS1 .GE. TT2) THEN
    PRINT*, ' '
    PRINT*, 'IT IS IMPOSSIBLE TO HAVE THE TUBE FLUID BEING'
    PRINT*, 'COOLED AND THE SHELL FLUID ENTRANCE TEMPERATURE'
    PRINT*, 'GREATER THAN THE TUBE FLUID EXIT TEMPERATURE. TRY'
    PRINT*, 'INPUTTING THE SHELL FLUID ENTRANCE TEMPERATURE'
    PRINT*, 'AGAIN.'
    PRINT*, ' '
    GO TO 713
END IF

IF (AGAIN .EQ. 'Y') THEN
    GO TO 714
END IF

PRINT*, ' '
715 PRINT*, 'WHAT IS THE FLOW RATE OF THE SHELL FLUID IN'
PRINT*, 'GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)'
PRINT*, ' '

READ*, MS

```

```

READ*, PICKFS

IF (AGAIN .EQ. 'Y') THEN
    GO TO 708
END IF

*** THE TEMPERATURES AND FLOW RATES ARE NOW INPUT BY THE
*** USER AS PROMPTED BY THE PROGRAM

PRINT*, ' '
709 PRINT*, 'WHAT IS THE ENTERING TEMPERATURE OF THE TUBE'
PRINT*, 'FLUID IN DEGREES F ? (ie 250.0, 366.5 etc.)'
PRINT*, ' '

READ*, TT1

PRINT*, ' '
PRINT*, 'WHAT IS THE EXIT TEMPERATURE OF THE TUBE FLUID'
PRINT*, 'IN DEGREES F ?'
PRINT*, ' '

READ*, TT2

*** CHECK TO SEE IF THE HEAT EXCHANGER TEMPERATURE DATA IS
*** ACCEPTABLE

IF (TT1 .EQ. TT2) THEN
    PRINT*, ' '
    PRINT*, 'THE ENTRANCE AND EXIT TEMPERATURES OF THE'
    PRINT*, 'TUBE FLUID CANNOT BE EQUAL. PLEASE TRY AGAIN.'
    PRINT*, ' '
    GO TO 709
END IF

*** CALCULATES THE AVERAGE TUBE-SIDE TEMPERATURE TO BE USED
*** IN CALCULATING THE FLUID PROPERTIES

TTAVE = (TT1 + TT2) / 2.0

IF (AGAIN .EQ. 'Y') THEN
    GO TO 710
END IF

PRINT*, ' '
711 PRINT*, 'WHAT IS THE FLOW RATE OF THE TUBE FLUID IN'
PRINT*, 'GALLONS PER MINUTE ? (ie 400.0, 385.7 etc.)'
PRINT*, ' '

READ*, MT

IF (AGAIN .EQ. 'Y') THEN
    GO TO 712
END IF

```

GO TO 704
END IF

*** THE USER IS ASKED TO PICK THE TYPE OF FLUIDS WANTED IN
*** THE SHELL AND IN THE TUBE OF THE EXCHANGER

PRINT*, ' '
705 PRINT*, 'WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN'
PRINT*, 'THE TUBES OF THE EXCHANGER? THE FOLLOWING'
PRINT*, 'FLUID CHOICES ARE AVAILABLE:'
PRINT*, ' '

| PRINT*, 'CHOICE NUMBER | FLUID NAME' |
|------------------------|-------------------------|
| PRINT*, ' 1.0 | WATER' |
| PRINT*, ' 2.0 | METHYL ALCOHOL' |
| PRINT*, ' 3.0 | GASOLINE' |
| PRINT*, ' 4.0 | DOWTHERM A' |
| PRINT*, ' 5.0 | KEROSENE' |
| PRINT*, ' 6.0 | SAE 10 LUBRICATING OIL' |
| PRINT*, ' 7.0 | ETHYLENE GLYCOL' |
| PRINT*, ' ' | |

PRINT*, 'PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED '
PRINT*, 'TUBE FLUID. (ie 1.0 IF THE FLUID IS WATER)'
PRINT*, ' '

READ*, PICKFT

IF (AGAIN .EQ. 'Y') THEN
GO TO 706
END IF

PRINT*, ' '
707 PRINT*, 'WHAT TYPE OF FLUID DO YOU WANT TO BE USED IN '
PRINT*, 'THE SHELL OF THE EXCHANGER? THE FOLLOWING'
PRINT*, 'FLUID CHOICES ARE AVAILABLE:'
PRINT*, ' '

| PRINT*, 'CHOICE NUMBER | FLUID NAME' |
|------------------------|-------------------------|
| PRINT*, ' 1.0 | WATER' |
| PRINT*, ' 2.0 | METHYL ALCOHOL' |
| PRINT*, ' 3.0 | GASOLINE' |
| PRINT*, ' 4.0 | DOWTHERM A' |
| PRINT*, ' 5.0 | KEROSENE' |
| PRINT*, ' 6.0 | SAE 10 LUBRICATING OIL' |
| PRINT*, ' 7.0 | ETHYLENE GLYCOL' |
| PRINT*, ' ' | |

PRINT*, 'PLEASE ENTER THE CHOICE NUMBER OF THE DESIRED '
PRINT*, 'SHELL FLUID. (ie 1.0 IF THE FLUID IS WATER)'
PRINT*, ' '

```

DTOT = DTO(I)
DTIN = DTI(I)
ATIN = ATI(I)
ATSU = ATS(I)
TPH = TP(I)
PI = 3.1415927

```

```

IF (AGAIN .EQ. 'Y') THEN
    GO TO 702
END IF

```

*** MATERIAL SPECIFICATIONS OF THE TUBING IS MADE

```

703 PRINT*, ' '
    PRINT*, 'WHAT MATERIAL SPECIFICATIONS DO YOU WANT FOR'
    PRINT*, 'THE TUBING ?'
    PRINT*, ' '
    PRINT*, 'CHOICE NUMBER          MATERIAL TYPE '
    PRINT*, ' '
    PRINT*, '      1.0          ALUMINUM TYPE 1100'
    PRINT*, '      2.0          ALUMINUM TYPE 3003'
    PRINT*, '      3.0          ALUMINUM TYPE 3004'
    PRINT*, '      4.0          ALUMINUM TYPE 6061'
    PRINT*, '      5.0          ALUMINUM TYPE 6063'
    PRINT*, '      6.0          CARBON STEEL'
    PRINT*, '      7.0          CARBON MOLY STEEL'
    PRINT*, '      8.0          CHROME MOLY STEEL (1%)'
    PRINT*, '      9.0          CHROME MOLY STEEL (2.25%)'
    PRINT*, '     10.0          CHROME MOLY STEEL (5%)'
    PRINT*, '     11.0          CHROME MOLY STEEL (12%)'
    PRINT*, '     12.0          STAINLESS STEEL (18%)'
    PRINT*, '     13.0          STAINLESS STEEL (25%)'
    PRINT*, '     14.0          ADMIRALTY METAL'
    PRINT*, '     15.0          COPPER'
    PRINT*, '     16.0          NICKEL'
    PRINT*, '     17.0          ZIRCONIUM'
    PRINT*, '     18.0          TITANIUM'
    PRINT*, ' '

```

```

PRINT*, 'PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF '
PRINT*, 'MATERIAL DESIRED.(ie 1.0 6.0 18.0 etc.)'
PRINT*, ' '

```

```

READ*, PICKM

```

```

I = INT(PICKM)

```

*** MATERIAL PROPERTIES ARE SET ONCE A CHOICE IS MADE

```

KW = KM(I)

```

```

CF = CFM(I)

```

```

IF (AGAIN .EQ. 'Y') THEN

```



```

PRINT*,
PRINT*, CHOICE NUMBER  OUTSIDE DIAMETER(IN)  BWG #
PRINT*,
PRINT*,      1.0          0.75          10'
PRINT*,      2.0          0.75          11'
PRINT*,      3.0          0.75          12'
PRINT*,      4.0          0.75          13'
PRINT*,      5.0          0.75          14'
PRINT*,      6.0          0.75          15'
PRINT*,      7.0          0.75          16'
PRINT*,      8.0          0.75          17'
PRINT*,      9.0          0.75          18'
PRINT*,     10.0          0.75          19'
PRINT*,     11.0          0.75          20'
PRINT*,
PRINT*,     12.0          1.0           8'
PRINT*,     13.0          1.0          10'
PRINT*,     14.0          1.0          11'
PRINT*,     15.0          1.0          12'
PRINT*,     16.0          1.0          13'
PRINT*,     17.0          1.0          14'
PRINT*,     18.0          1.0          15'
PRINT*,     19.0          1.0          16'
PRINT*,     20.0          1.0          18'
PRINT*,     21.0          1.0          20'
PRINT*,
PRINT*,     22.0          1.25          7'
PRINT*,     23.0          1.25          8'
PRINT*,     24.0          1.25          10'
PRINT*,     25.0          1.25          11'
PRINT*,     26.0          1.25          12'
PRINT*,     27.0          1.25          13'
PRINT*,     28.0          1.25          14'
PRINT*,     29.0          1.25          16'
PRINT*,     30.0          1.25          17'
PRINT*,     31.0          1.25          18'
PRINT*,     32.0          1.25          20'
PRINT*,
PRINT*,     33.0          1.5           10'
PRINT*,     34.0          1.5           12'
PRINT*,     35.0          1.5           14'
PRINT*,     36.0          1.5           16'
PRINT*,

```

```

PRINT*, PLEASE ENTER THE CHOICE NUMBER OF THE TYPE OF
PRINT*, TUBING DESIRED.(ie 1.0 6.0 34.0 etc.)
PRINT*,

```

```

READ*, PICKT

```

```

I = INT(PICKT)

```

```

***  TUBE DIMENSION VARIABLES ARE SET ONCE A CHOICE IS MADE

```

\$ 0.606,0.620,0.634,0.652,0.666,0.680,0.670,0.732,
 \$ 0.760,0.782,0.810,0.834,0.856,0.870,0.902,0.930,
 \$ 0.890,0.920,0.982,1.010,1.032,1.060,1.084,1.120,
 \$ 1.134,1.152,1.180,1.232,1.282,1.334,1.370/

DATA (TF(I), I=1,34)/1.152,1.008,0.864,0.720,0.360,
 \$ 0.2448,0.1728,0.144,0.12672,0.1008,0.0720,0.06048,
 \$ 0.05328,0.0504,0.04752,0.04464,0.0432,0.04176,
 \$ 0.04032,0.0396,0.03312,0.02952,0.02736,0.025776,
 \$ 0.02448,0.02304,0.02232,0.0216,0.021456,0.018,
 \$ 0.01656,0.015552,0.014688,0.012672/

DATA (TRE(I), I=1,34)/60.0,70.0,80.0,100.0,200.0,300.0,
 \$ 400.0,500.0,600.0,700.0,1000.0,2000.0,3000.0,4000.0,
 \$ 5000.0,6000.0,7000.0,8000.0,9000.0,10000.0,20000.0,
 \$ 30000.0,40000.0,50000.0,60000.0,70000.0,80000.0,
 \$ 90000.0,100000.0,200000.0,300000.0,400000.0,
 \$ 500000.0,1000000.0/

DATA (SF(I), I=1,34)/1.4112,0.8928,0.6912,0.6048,
 \$ 0.5472,0.5323,0.5184,0.504,0.4968,0.4896,0.4752,
 \$ 0.4176,0.3816,0.36,0.3456,0.3384,0.3312,0.32544,
 \$ 0.31392,0.3024,0.2664,0.2448,0.23184,0.2232,0.216,
 \$ 0.21024,0.20304,0.2016,0.19872,0.1728,0.1584,0.1512,
 \$ 0.144,0.1296/

DATA (SRE(I), I=1,34)/50.0,100.0,200.0,300.0,400.0,
 \$ 500.0,600.0,700.0,800.0,900.0,1000.0,2000.0,3000.0,
 \$ 4000.0,5000.0,6000.0,7000.0,8000.0,9000.0,10000.0,
 \$ 20000.0,30000.0,40000.0,50000.0,60000.0,70000.0,
 \$ 80000.0,90000.0,100000.0,200000.0,300000.0,400000.0,
 \$ 500000.0,1000000.0/

DATA (TP(I), I=1,11)/11*0.9375/
 DATA (TP(I), I=12,21)/10*1.25/
 DATA (TP(I), I=22,32)/11*1.5625/
 DATA (TP(I), I=33,36)/4*1.875/

DATA (KM(I), I=1,18)/123.0,96.0,97.0,95.0,116.0,30.0,
 \$ 29.0,27.0,25.0,21.0,14.0,9.3,7.8,70.0,225.0,38.0,
 \$ 12.0,11.3/

DATA (CFM(I), I=1,18)/0.7237,0.7389,0.7389,0.7542,0.7542,
 \$ 0.0778,0.0957,0.1221,0.2188,0.2977,0.5435,0.7542,
 \$ 1.00,0.8235,0.8569,2.2443,3.00,3.00/

*** THE USER PICKS THE SIZE OF TUBING HE WANTS FOR THE TUBES
 *** OF HIS HEAT EXCHANGER. THESE COMMON SIZES OF TUBING AND
 *** ASSOCIATED DIMENSIONS WERE TAKEN FROM THE PATTERSON-
 *** KELLEY BOOK ON HEAT EXCHANGERS.

4 PRINT*, ' WHAT SIZE OF TUBING DO YOU WANT TO USE IN '
 PRINT*, 'YOUR HEAT EXCHANGER? (REMEMBER THE SMALLER THE '
 PRINT*, 'BWG NUMBER THE THICKER THE WALL OF THE TUBING)'

```

PRINT*, '      INPUT DATA                UNITS'
PRINT*, '
PRINT*, 'TUBE FLUID ENTRANCE TEMP        DEGREES F'
PRINT*, 'TUBE FLUID EXIT TEMP            DEGREES F'
PRINT*, 'TUBE FLUID FLOW RATE             GALLONS PER MINUTE'
PRINT*, 'SHELL FLUID ENTRANCE TEMP        DEGREES F'
PRINT*, 'SHELL FLUID FLOW RATE            GALLONS PER MINUTE'
PRINT*, 'MAXIMUM TUBE LENGTH              FEET'
PRINT*, 'MAX TUBE PRESSURE LOSS           POUNDS PER SQUARE INCH'
PRINT*, 'MAX SHELL PRESSURE LOSS          POUNDS PER SQUARE INCH'
PRINT*, '

```

```

PRINT*, 'WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.'
READ (5,101) ACALC

```

```

PRINT*, '      THE MAJOR SOURCES OF INFORMATION USED TO '
PRINT*, 'FORMULATE THIS PROGRAM ARE:'
PRINT*, '
PRINT*, 'BREESE, J. - SHELL AND TUBE HEAT EXCHANGERS'
PRINT*, 'KAYS & LONDON - COMPACT HEAT EXCHANGERS'
PRINT*, 'KERN, D. - PROCESS HEAT TRANSFER'
PRINT*, 'HOLMAN, J.P. - HEAT TRANSFER'
PRINT*, 'CHASE, T.A. - PERSONAL COMMUNICATION'
PRINT*, 'FANARITIS, J.P. - HEAT EXCHANGER TECHNOLOGY'
PRINT*, 'GAGGIOLI, R.A. - EFFICIENCY AND COSTING'
PRINT*, 'PATTERSON-KELLEY CO. - HEAT EXCHANGERS'
PRINT*, 'TEMA - STANDARDS OF TUBULAR EXCHANGERS'
PRINT*, '

```

```

PRINT*, 'WHEN YOU ARE READY TO PROCEED TYPE Y <RET>.'
READ (5,101) ACALC

```

```

DATA (DSI(I), I=1,28)/6.0,8.0,10.0,12.0,14.0,16.0,18.0,
$ 20.0,22.0,24.0,26.0,28.0,30.0,32.0,34.0,36.0,38.0,
$ 40.0,42.0,44.0,46.0,48.0,50.0,52.0,54.0,56.0,58.0,
$ 60.0/

```

```

DATA (DTO(I), I=1,11)/11*0.75/
DATA (DTO(I), I=12,21)/10*1.0/
DATA (DTO(I), I=22,32)/11*1.25/
DATA (DTO(I), I=33,36)/4*1.50/

```

```

DATA (ATI(I), I=1,36)/0.1822,0.2043,0.223,0.247,0.268,
$ 0.289,0.302,0.314,0.334,0.348,0.363,0.355,0.421,
$ 0.455,0.479,0.515,0.546,0.576,0.594,0.639,0.679,
$ 0.622,0.662,0.757,0.801,0.838,0.882,0.923,0.985,
$ 1.012,1.042,1.092,1.192,1.291,1.397,1.474/

```

```

DATA (ATS(I), I=1,11)/11*0.1963/
DATA (ATS(I), I=12,21)/10*0.2618/
DATA (ATS(I), I=22,32)/11*0.3272/
DATA (ATS(I), I=33,36)/4*0.3927/

```

```

DATA (DTI(I), I=1,36)/0.482,0.510,0.532,0.560,0.584,

```

C PROGRAM MAIN1 LAST UPDATED: 5 MARCH 85
C
C NICHOLAS J. KOLAR
C MECHANICAL ENGINEERING MASTER'S THESIS
C DESIGN OF SHELL AND TUBE HEAT EXCHANGERS
C JANUARY 1985

DIMENSION DSI(28),QTL8(28),QTH8(28),QTL6(28),
\$ QTH6(28),QTL4(28),QTH4(28),QTL2(28),QTH2(28),
\$ QSK1L(28),QSK1H(28),QSK2L(28),QSK2H(28),QSK3L(28),
\$ QSK3H(28),TF(34),TRE(34),SF(34),SRE(34),TP(36),
\$ DTO(36),ATI(36),ATS(36),DTI(36),DSIU(28),NT2(28),
\$ NT4(28),NT6(28),NT8(28),AL2(28),AL4(28),AL6(28),
\$ AL8(28),KM(18),CFM(18)

CHARACTER*1 ACALC,ALENTH,AFOUL,AGAIN,ATUBE,AMATE,ATFLU,
\$ ASFLU,ATTEM,ATGPM,ASTEM,ASGPM,AFFAC,ALREST,
\$ APRES
REAL NTU ,MT,MS,LRESTR,L,NP,K,KS,KT,K1,KM,KW

PRINT*, ' THIS PROGRAM CALCULATES THE REQUIRED LENGTH,
PRINT*, ' SIZE OF SHELL, BAFFLE SPACING, NUMBER OF TUBE
PRINT*, ' PASSES, AND ESTIMATED CAPITAL COST OF A TEMA
PRINT*, ' TYPE E SHELL AND TUBE HEAT EXCHANGER WITH 2, 4,
PRINT*, ' 6, OR 8 TUBE PASSES. YHIS PROGRAM ALSO PROVIDES
PRINT*, ' CALCULATIONS OF THE SHELL-SIDE AND TUBE-SIDE
PRINT*, ' PRESSURE DROPS. THE USER PICKS THE SIZE OF THE
PRINT*, ' TUBING TO BE USED ALONG WITH THE TUBE MATERIAL.
PRINT*, ' THE USER SETS THE MAXIMUM LENGTH ALONG WITH THE
PRINT*, ' MAXIMUM ALLOWABLE PRESSURE DROPS ON THE SHELL
PRINT*, ' AND TUBE SIDES.'
PRINT*, '
PRINT*, ' THE PROGRAM OUTPUTS THE EXCHANGER CONFIGURATIONS
PRINT*, ' THAT WILL MEET THE PROBLEM LIMITATIONS. FINAL
PRINT*, ' DESIGN DATA ON ANY CONFIGURATION IS PROVIDED AS
PRINT*, ' REQUESTED. ALTERNATE EXCHANGER CONFIGURATIONS
PRINT*, ' CAN BE COMPARED, AND ORIGINAL INPUT DATA CAN BE
PRINT*, ' CHANGED TO DETERMINE THEIR EFFECTS ON THE FINAL
PRINT*, ' DESIGN DATA.'
PRINT*, '
PRINT*, ' THIS PROGRAM IS LIMITED TO LIQUID-TO-LIQUID
PRINT*, ' APPLICATIONS UNDER LOW PRESSURE CONDITIONS (<300 PSI).
PRINT*, '
PRINT*, ' A LIMITED NUMBER OF COMMON FLUIDS
PRINT*, ' ARE AVAILABLE TO BE CHOSEN AS FLUIDS IN THE
PRINT*, ' TUBE AND SHELL. THIS PROGRAM CAN BE EASILY
PRINT*, ' MODIFIED TO ACCEPT ADDITIONAL FLUIDS FOR WHICH
PRINT*, ' DENSITY, VISCOSITY, THERMAL CONDUCTIVITY AND SPECIFIC
PRINT*, ' HEAT DATA ARE AVAILABLE.'
PRINT*, '
PRINT*, ' THE USER MUST INPUT THE FOLLOWING DATA IN
PRINT*, ' THE UNITS SPECIFIED.'
PRINT*, '

APPENDIX B
MAIN PROGRAM LISTING

| <u>Variable Name</u> | <u>Explanation</u> | <u>Units</u> |
|----------------------|--|---|
| TSAVE | Average temperature of the shell fluid. | °F |
| TS1 | Entrance temperature of the shell fluid. | °F |
| TS2 | Exit temperature of the shell fluid. | °F |
| TS2NEW | Adjusted exit temperature of the shell fluid. | °F |
| TTAVE | Average temperature of the tube fluid. | °F |
| TT1 | Entrance temperature of the tube fluid. | °F |
| TT2 | Exit temperature of the tube fluid. | °F |
| U | Average overall heat transfer coefficient of the exchanger (estimated). | $\frac{\text{BTU}}{\text{HR-FT}^2\text{-}^\circ\text{F}}$ |
| UC | Average overall heat transfer coefficient of exchanger with clean tubes. | $\frac{\text{BTU}}{\text{HR-FT}^2\text{-}^\circ\text{F}}$ |
| UCALC | Overall heat transfer coefficient of the exchanger (calculated). | $\frac{\text{BTU}}{\text{HR-FT}^2\text{-}^\circ\text{F}}$ |
| VS | Shell fluid velocity. | FT/SEC |
| VT | Tube fluid velocity. | FT/SEC |
| Z | Estimated capital cost of the exchanger. | \$ |

END IF

IF(PICKFT .EQ. 6.0) THEN
CALL DSAE(TAVE,DEN)
DENT = DEN
CALL CPSAE(TAVE,CP)
CPT = CP
CALL VSAE(TAVE,DV)
DVT = DV
CALL KSAE(TAVE,K)
KT = K

END IF

IF(PICKFT .EQ. 7.0) THEN
CALL DEGLY(TAVE,DEN)
DENT = DEN
CALL CPEGLY(TAVE,CP)
CPT = CP
CALL VEGLY(TAVE,DV)
DVT = DV
CALL KEGLY(TAVE,K)
KT = K

END IF

*** CALCULATE THE SHELL FLUID EXIT TEMPERATURE USING THE
*** HEAT BALANCE EQUATION. NOTE THAT 60.0 IS A CONVERSION
*** FOR MIN/HR AND 7.4805 IS A CONVERSION FOR GALLONS/
*** CUBIC FEET.

IF (CASE .EQ. 1.0) THEN
Q = (MT * DENT * CPT * 60.0 / 7.4805) * (TT2-TT1)
END IF

IF (CASE .EQ. 2.0) THEN
Q = (MT * DENT * CPT * 60.0 / 7.4805) * (TT1-TT2)
END IF

*** GET AN AVERAGE VALUE FOR DENSITY AND SPECIFIC HEAT FOR
*** INITIAL TRIAL AND ERROR GUESS

IF(PICKFS .EQ. 1.0) THEN
DENS = 62.4
CPS = 1.0
END IF

IF(PICKFS .EQ. 2.0) THEN
DENS = 48.1
CPS = 0.62
END IF

IF(PICKFS .EQ. 3.0) THEN
DENS = 42.7
CPS = 0.565
END IF

```
IF(PICKFT .EQ. 1.0) THEN
  CALL DWATER(TAVE,DEN)
  DENT = DEN
  CALL CPWATR(TAVE,CP)
  CPT = CP
  CALL VWATER(TAVE,DV)
  DVT = DV
  CALL KWATER(TAVE,K)
  KT = K
END IF
```

```
IF(PICKFT .EQ. 2.0) THEN
  CALL DMETH(TAVE,DEN)
  DENT = DEN
  CALL CPMETH(TAVE,CP)
  CPT = CP
  CALL VMETH(TAVE,DV)
  DVT = DV
  CALL KMETH(TAVE,K)
  KT = K
END IF
```

```
IF(PICKFT .EQ. 3.0) THEN
  CALL DGAS(TAVE,DEN)
  DENT = DEN
  CALL CPGAS(TAVE,CP)
  CPT = CP
  CALL VGAS(TAVE,DV)
  DVT = DV
  CALL KGAS(TAVE,K)
  KT = K
END IF
```

```
IF(PICKFT .EQ. 4.0) THEN
  CALL DDOWA(TAVE,DEN)
  DENT = DEN
  CALL CPDOWA(TAVE,CP)
  CPT = CP
  CALL VDOWA(TAVE,DV)
  DVT = DV
  CALL KDOWA(TAVE,K)
  KT = K
END IF
```

```
IF(PICKFT .EQ. 5.0) THEN
  CALL DKERO(TAVE,DEN)
  DENT = DEN
  CALL CPKERO(TAVE,CP)
  CPT = CP
  CALL VKERO(TAVE,DV)
  DVT = DV
  CALL KKERO(TAVE,K)
  KT = K
```



```
IF(PICKFS .EQ. 4.0) THEN
  DENS = 56.8
  CPS = 0.60
END IF
```

```
IF(PICKFS .EQ. 5.0) THEN
  DENS = 44.4
  CPS = 0.60
END IF
```

```
IF(PICKFS .EQ. 6.0) THEN
  DENS = 52.25
  CPS = 0.52
END IF
```

```
IF(PICKFS .EQ. 7.0) THEN
  DENS = 66.2
  CPS = 0.644
END IF
```

```
*** SOLVE HEAT BALANCE EQUATION FOR INITIAL GUESS OF SHELL
*** FLUID EXIT TEMPERATURE
```

```
IF (CASE .EQ. 1.0) THEN
  TS2 = TS1 - (Q/ ( MS * DENS * CPS * 60.0 / 7.4805))
END IF
```

```
IF (CASE .EQ. 2.0) THEN
  TS2 = TS1 + (Q/ ( MS * DENS * CPS * 60.0 / 7.4805))
END IF
```

```
*** CALCULATE THE SHELL FLUID PROPERTIES BASED UPON THE
*** AVERAGE TEMPERATURE OF THE SHELL FLUID
```

```
48 TSAVE = (TS2+TS1) / 2.0
TAVE = TSAVE
```

```
IF(PICKFS .EQ. 1.0) THEN
  CALL DWATER(TAVE,DEN)
  DENS = DEN
  CALL CPWATR(TAVE,CP)
  CPS = CP
  CALL VWATER(TAVE,DV)
  DVS = DV
  CALL KWATER(TAVE,K)
  KS = K
END IF
```

```
IF(PICKFS .EQ. 2.0) THEN
  CALL DMETH(TAVE,DEN)
  DENS = DEN
  CALL CPMETH(TAVE,CP)
  CPS = CP
```

```
CALL VMETH(TAVE,DV)
DVS = DV
CALL KMETH(TAVE,K)
KS = K
END IF
```

```
IF(PICKFS .EQ. 3.0) THEN
CALL DGAS(TAVE,DEN)
DENS = DEN
CALL CPGAS(TAVE,CP)
CPS = CP
CALL VGAS(TAVE,DV)
DVS = DV
CALL KGAS(TAVE,K)
KS = K
END IF
```

```
IF(PICKFS .EQ. 4.0) THEN
CALL DDOWA(TAVE,DEN)
DENS = DEN
CALL CPDOWA(TAVE,CP)
CPS = CP
CALL VDOWA(TAVE,DV)
DVS = DV
CALL KDOWA(TAVE,K)
KS = K
END IF
```

```
IF(PICKFS .EQ. 5.0) THEN
CALL DKERO(TAVE,DEN)
DENS = DEN
CALL CPKERO(TAVE,CP)
CPS = CP
CALL VKERO(TAVE,DV)
DVS = DV
CALL KKERO(TAVE,K)
KS = K
END IF
```

```
IF(PICKFS .EQ. 6.0) THEN
CALL DSAE(TAVE,DEN)
DENS = DEN
CALL CPSAE(TAVE,CP)
CPS = CP
CALL VSAE(TAVE,DV)
DVS = DV
CALL KSAE(TAVE,K)
KS = K
END IF
```

```
IF(PICKFS .EQ. 7.0) THEN
CALL DEGLY(TAVE,DEN)
DENS = DEN
CALL CPEGLY(TAVE,CP)
```

```

      CPS = CP
      CALL VEGLY(TAVE,DV)
      DVS = DV
      CALL KEGLY(TAVE,K)
      KS = K
    END IF

***  RECHECK SHELL EXIT TEMPERATURE CALCULATION USING NEW
***  VALUES FOR DENSITY AND SPECIFIC HEAT OF THE SHELL FLUID

    IF (CASE .EQ. 1.0) THEN
      TS2NEW = TS1 - (Q / ( MS * DENS * CPS * 60.0 / 7.4805))
    END IF

    IF (CASE .EQ. 2.0) THEN
      TS2NEW = TS1 + (Q / ( MS * DENS * CPS * 60.0 / 7.4805))
    END IF

    IF (ABS (TS2-TS2NEW) .GT. 1.0) THEN
      TS2 = TS2NEW
      GO TO 48
    END IF

***  DETERMINE WHICH FLUID HAS THE MAXIMUM CAPACITY RATE FOR
***  USE IN THE NTU METHOD

    CS = MS * DENS * CPS * 60.0 / 7.4805
    CT = MT * DENT * CPT * 60.0 / 7.4805

    IF (CS .GE. CT) THEN
      CMIN = CT
      T1MIN = TT1
      T2MIN = TT2
      CMAX = CS
      T1MAX = TS1
      T2MAX = TS2
    ELSE
      CMIN = CS
      T1MIN = TS1
      T2MIN = TS2
      CMAX = CT
      T1MAX = TT1
      T2MAX = TT2
    END IF

***  CALCULATE THE HEAT TRANSFER EFFECTIVENESS

    EFF = ABS(T2MIN -T1MIN) / ABS(T1MAX-T1MIN)

***  CHECK FOR IMPOSSIBLE CONDITIONS

    IF (EFF .GT. 1.0) THEN
      PRINT*, ' '

```

```

PRINT*, 'THE TEMPERATURES PROVIDED AS INPUT ARE SUCH'
PRINT*, 'THAT THE HEAT EXCHANGER WOULD BE REQUIRED TO'
PRINT*, 'ACCOMPLISH AN IMPOSSIBLE FEAT. THE MAXIMUM'
PRINT*, 'POSSIBLE HEAT TRANSFER FOR AN EXCHANGER OF'
PRINT*, 'INFINITE LENGTH IS LESS THAN WHAT YOU REQUIRE'
PRINT*, 'THIS HEAT EXCHANGER TO ACCOMPLISH. TRY'
PRINT*, 'INPUTTING THE TEMPERATURES ONCE AGAIN.'
PRINT*, ' '
GO TO 709
END IF

```

```

RATIO = CMIN / CMAX

```

```

*** CALCULATE THE NUMBER OF HEAT TRANSFER UNITS FOR THE
*** HEAT EXCHANGER USING EQUATION FROM HOLMAN'S - HEAT
*** TRANSFER ON PAGE 457

```

```

NTU = ( -(1.0 + (RATIO **2.0))**(-0.5)) * LOG (((2.0/EFF) - 1.0
$ - RATIO - ((1.0 + (RATIO **2.0))**0.5)) /
$ ((2.0/EFF) - 1.0 - RATIO + ((1.0 + (RATIO ** 2.0))**0.5)))

```

```

*** CALCULATE THE OVERALL HEAT TRANSFER COEFFICIENT USING
*** CLEAN TUBING. THE VALUES UTILIZED IN THIS PROGRAM WERE
*** TAKEN FROM HEAT EXCHANGERS THERMAL-HYDRAULIC
*** FUNDAMENTALS AND DESIGN BY KAKAC AND BERGLES.
*** VALUES FOR FOULING RESISTANCES ARE TAKEN FROM TEMA
*** STANDARDS HANDBOOK P 140. UNITS ARE HR-SQ FT-DEG F/BTU

```

```

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 1.0) THEN
    UC = 275.0
    RFOULT = .0015
    RO = .00075
    RI = .00075

```

```

END IF

```

```

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 2.0) THEN
    UC = 150.0
    RFOULT = .0015
    RO = .00075
    RI = .00075

```

```

END IF

```

```

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 3.0) THEN
    UC = 150.0
    RFOULT = .0015
    RO = .00075
    RI = .00075

```

```

END IF

```

```

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 4.0) THEN
    UC = 100.0
    RFOULT = .002
    RO = .00125
    RI = .00075

```

END IF

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 5.0) THEN
UC = 100.0
RFOULT = .002
RO = .00125
RI = .00075

END IF

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 6.0) THEN
UC = 57.5
RFOULT = .0025
RO = .00175
RI = .00075

END IF

IF (PICKFT .EQ. 1.0 .AND. PICKFS .EQ. 7.0) THEN
UC = 57.5
RFOULT = .0025
RO = .00175
RI = .00075

END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 1.0) THEN
UC = 150.0
RFOULT = .0015
RO = .00075
RI = .00075

END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 2.0) THEN
UC = 115.0
RFOULT = .002
RO = .001
RI = .001

END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 3.0) THEN
UC = 115.0
RFOULT = .002
RO = .001
RI = .001

END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 4.0) THEN
UC = 85.0
RFOULT = .0025
RO = .0015
RI = .001

END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 5.0) THEN
UC = 85.0
RFOULT = .0025

```
      RO = .0015
      RI = .001
END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 6.0) THEN
      UC = 50.0
      RFOULT = .003
      RO = .002
      RI = .001
END IF

IF (PICKFT .EQ. 2.0 .AND. PICKFS .EQ. 7.0) THEN
      UC = 50.0
      RFOULT = .003
      RO = .002
      RI = .001
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 1.0) THEN
      UC = 150.0
      RFOULT = .0015
      RO = .00075
      RI = .00075
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 2.0) THEN
      UC = 115.0
      RFOULT = .002
      RO = .001
      RI = .001
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 3.0) THEN
      UC = 115.0
      RFOULT = .002
      RO = .001
      RI = .001
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 4.0) THEN
      UC = 85.0
      RFOULT = .0025
      RO = .0015
      RI = .001
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 5.0) THEN
      UC = 85.0
      RFOULT = .0025
      RO = .0015
      RI = .001
END IF

IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 6.0) THEN
```

```

        UC = 50.0
        RFOULT = .003
        RO = .002
        RI = .001
    END IF

    IF (PICKFT .EQ. 3.0 .AND. PICKFS .EQ. 7.0) THEN
        UC = 50.0
        RFOULT = .003
        RO = .002
        RI = .001
    END IF

    IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 1.0) THEN
        UC = 100.0
        RFOULT = .002
        RO = .00075
        RI = .00125
    END IF

    IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 2.0) THEN
        UC = 85.0
        RFOULT = .0025
        RO = .00125
        RI = .00125
    END IF

    IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 3.0) THEN
        UC = 85.0
        RFOULT = .0025
        RO = .00125
        RI = .00125
    END IF

    IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 4.0) THEN
        UC = 65.0
        RFOULT = .003
        RO = .00175
        RI = .00125
    END IF

    IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 5.0) THEN
        UC = 65.0
        RFOULT = .003
        RO = .00175
        RI = .00125
    END IF

    IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 6.0) THEN
        UC = 35.0
        RFOULT = .0035
        RO = .00225
        RI = .00125
    END IF

```

```
IF (PICKFT .EQ. 4.0 .AND. PICKFS .EQ. 7.0) THEN
  UC = 35.0
  RFOULT = .0035
  RO = .00225
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 1.0) THEN
  UC = 100.0
  RFOULT = .002
  RO = .00075
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 2.0) THEN
  UC = 85.0
  RFOULT = .0025
  RO = .00125
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 3.0) THEN
  UC = 85.0
  RFOULT = .0025
  RO = .00125
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 4.0) THEN
  UC = 65.0
  RFOULT = .003
  RO = .00175
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 5.0) THEN
  UC = 65.0
  RFOULT = .003
  RO = .00175
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 6.0) THEN
  UC = 35.0
  RFOULT = .0035
  RO = .00225
  RI = .00125
```

```
END IF
```

```
IF (PICKFT .EQ. 5.0 .AND. PICKFS .EQ. 7.0) THEN
  UC = 35.0
  RFOULT = .0035
  RO = .00225
```



```

        RI = .00125
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 1.0) THEN
    UC = 57.5
    RFOULT = .0025
    RO = .00175
    RI = .00075
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 2.0) THEN
    UC = 50.0
    RFOULT = .003
    RO = .001
    RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 3.0) THEN
    UC = 50.0
    RFOULT = .003
    RO = .001
    RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 4.0) THEN
    UC = 35.0
    RFOULT = .0035
    RO = .0015
    RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 5.0) THEN
    UC = 35.0
    RFOULT = .0035
    RO = .0015
    RI = .002
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 6.0) THEN
    UC = 20.0
    RFOULT = .005
    RO = .0025
    RI = .0025
END IF

IF (PICKFT .EQ. 6.0 .AND. PICKFS .EQ. 7.0) THEN
    UC = 20.0
    RFOULT = .005
    RO = .0025
    RI = .0025
END IF

IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 1.0) THEN

```

```

        UC = 57.5
        RFOULT = .0025
        RO = .00175
        RI = .00075
    END IF

    IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 2.0) THEN
        UC = 50.0
        RFOULT = .003
        RO = .001
        RI = .002
    END IF

    IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 3.0) THEN
        UC = 50.0
        RFOULT = .003
        RO = .001
        RI = .002
    END IF

    IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 4.0) THEN
        UC = 35.0
        RFOULT = .0035
        RO = .0015
        RI = .002
    END IF

    IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 5.0) THEN
        UC = 35.0
        RFOULT = .0035
        RO = .0015
        RI = .002
    END IF

    IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 6.0) THEN
        UC = 20.0
        RFOULT = .005
        RO = .0025
        RI = .0025
    END IF

    IF (PICKFT .EQ. 7.0 .AND. PICKFS .EQ. 7.0) THEN
        UC = 20.0
        RFOULT = .005
        RO = .0025
        RI = .0025
    END IF

```

```

*** THE USER IS ASKED IF FOULING RESISTANCE SHOULD BE
*** CONSIDERED IN THE EVALUATION OF THE OVERALL HEAT
*** TRANSFER COEFFICIENT.

```

```

    IF (AFFAC .EQ. 'N') THEN
        GO TO 50
    
```

```

END IF

PRINT*, ' '
PRINT*, 'DO YOU WANT TO ADD A FOULING FACTOR TO THE HEAT '
PRINT*, 'TRANSFER COEFFICIENT GIVEN FOR CLEAN TUBES ? '
PRINT*, 'ANSWER Y FOR YES AND N FOR NO.'
PRINT*, ' '
READ (5,101) AFOUL

50  IF (AFOUL .EQ. 'Y') THEN
      U = 1.0 / ( RFOULT + ( 1.0 / UC ) )
    ELSE
      U = UC
    END IF

***  DETERMINE WHAT LENGTH RESTRICTIONS ARE TO BE CONSIDERED
***  FOR THIS HEAT EXCHANGER APPLICATION

    IF (ALREST .EQ. 'N') THEN
      GO TO 51
    END IF

    PRINT*, ' '
    PRINT*, 'WHAT IS THE MAXIMUM LENGTH THAT THE HEAT '
    PRINT*, 'EXCHANGER TUBES CAN BE FOR THIS APPLICATION ? '
    PRINT*, 'REMEMBER, THE SHELL WILL HAVE TO BE REMOVED FOR '
    PRINT*, 'CLEANING SO THE MAXIMUM LENGTH OF THE HEAT '
    PRINT*, 'EXCHANGER TUBES SHOULD BE LESS THAN ONE HALF THE '
    PRINT*, 'LENGTH OF THE ROOM THAT THE HEAT EXCHANGER WILL '
    PRINT*, 'BE OPERATING IN. TYPE IN THE MAXIMUM LENGTH IN '
    PRINT*, 'FEET THAT THE HEAT EXCHANGER TUBES CAN BE.'
    PRINT*, '( ie 5.0 , 7.5, 19.6 etc.)'
    PRINT*, ' '

    READ*, LRESTR

51  IF (APRES .EQ. 'N') THEN
      GO TO 52
    END IF

***  DETERMINE WHAT PRESSURE LOSSES ARE ACCEPTABLE ON BOTH
***  THE SHELL AND THE TUBE SIDES OF THE EXCHANGER

    PRINT*, ' '
    PRINT*, 'WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN '
    PRINT*, 'THE TUBES IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)'
    PRINT*, ' '

    READ*, AMAXTP

    PRINT*, ' '
    PRINT*, 'WHAT IS THE MAXIMUM ALLOWABLE PRESSURE LOSS IN '
    PRINT*, 'THE SHELL IN POUNDS PER SQUARE INCH ? (ie 7.4 etc)'
    PRINT*, ' '

```

READ*, AMAXSP

*** NOW THAT THE TUBE FLOW RATE, THE SHELL FLOW RATE, THE
*** MAXIMUM TUBE LENGTH, AND THE MAXIMUM TUBE AND SHELL PRESSURE
*** LOSSES HAVE BEEN SPECIFIED, THE PROGRAM CHECKS EACH AVAILABLE
*** SHELL SIZE AND NUMBER OF TUBE PASSES TO SEE WHICH 1-2 HEAT
*** EXCHANGER CONFIGURATIONS MEET THE REQUIRED HEAT DUTY AND
*** LENGTH AND PRESSURE LIMITATIONS IMPOSED BY THE USER

*** THE FOLLOWING FIVE VARIABLES ARE USED AS COUNTERS IN THE
*** PRINTING ALGORITHM:

52 J = 0
J2 = 0
J4 = 0
J6 = 0
J8 = 0

PRINT*, ' '

*** ALL AVAILABLE 2-PASS EXCHANGER CONFIGURATIONS ARE CHECKED

DO 60, I = 1,28

K2 = 0
M1 = 0
M2 = 0
M3 = 0

*** FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
*** RECOMMENDED MINIMUMS

IF(MT .GE. QTL2(I) .AND. MT .LE. QTH2(I)) THEN
K2 = 1
END IF

IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
M1 = 1
END IF

IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
M2 = 1
END IF

IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
M3 = 1
END IF

IF(K2 .EQ. 0) THEN
GO TO 60
END IF

```

        WRITE(*,604) PICKS*DSI(N)
604   FORMAT ('          ',F6.2, ' INCHES')
        PRINT*,' '
        WRITE(*,605) L
605   FORMAT ('EXCHANGER TUBE LENGTH: ',F8.3, ' FEET')
        PRINT*,' '
        WRITE(*,606) PLTTOT
606   FORMAT ('TUBE-SIDE PRESSURE LOSS: ',F6.2,
$       ' POUNDS PER SQUARE INCH')
        PRINT*,' '
        WRITE(*,607) PLSTOT
607   FORMAT ('SHELL-SIDE PRESSURE LOSS: ',F6.2,
$       ' POUNDS PER SQUARE INCH')
        PRINT*,' '
        WRITE(*,608) Z
608   FORMAT ('ESTIMATED CAPITAL COST: $ ',F9.2)
        PRINT*,' '

***   THESE VALUES ARE PRINTED TO SEE HOW CLOSE THE COMPUTED
***   VALUE OF THE OVERALL HEAT TRANSFER COEFFICIENT COMES
***   TO THE AVERAGE VALUE FOUND IN THE LITERATURE

        PRINT*,'U = ', U
        PRINT*,'UCALC = ', UCALC

        PRINT*,' '
        PRINT*,'DO YOU WANT TO TRY A DIFFERENT HEAT EXCHANGER'
        PRINT*,'CONFIGURATION ? TYPE Y FOR YES AND N FOR NO.'
        PRINT*,' '
        READ (5,101) ACALC
101   FORMAT (1A1)

        IF(ACALC .EQ. 'N') THEN
            GO TO 700
        END IF

        GO TO 71

700   PRINT*,' '
        PRINT*,'DO YOU WANT TO CHANGE ANY OF THE INPUT PARAMETERS'
        PRINT*,'TO THIS PROBLEM ? TYPE Y FOR YES AND N FOR NO'
        PRINT*,' '

        READ (5,101) AGAIN

        IF (AGAIN .EQ. 'Y') THEN

            PRINT*,' '
            PRINT*,'DO YOU WANT TO CHANGE THE TUBING SIZE ?'
            PRINT*,' '

            READ (5,101) ATUBE

            IF (ATUBE .EQ. 'Y') THEN

```

*** DEPENDS ON THE REYNOLDS NUMBER USING SEIDER AND TATE
 *** FRICTION FACTOR CORRELATIONS FROM PAGE 836 OF KERN

CALL TFRICT(RET,TRE,TF,FRT)

*** CALCULATE THE PRESSURE LOSSES IN THE TUBE DUE TO BOTH
 *** THE FRICTION LOSSES AND THE RETURN LOSSES USING
 *** EQUATIONS FROM PAGE 148 AND 836 OF KERN. NOTE THAT
 *** .006944 IS A CONVERSION FACTOR FOR PSF TO PSI AND
 *** .4335 IS A CONVERSION FACTOR FOR FEET OF WATER TO PSI

TAVE = TTAVE
 CALL DWATER(TAVE,DEN)
 DENW = DEN

PLTF = ((FRT*NP*L*(GT**2.0)) / ((DTIN/12.0)* 2.0 * AG
 \$ * DENT * ((DVT/DVTF)**0.14))) * 0.006944

PLTR = ((2.0 * NP * (VT**2.0)) / (2.0 * AG *
 \$ (DENT/DENW))) * 0.4335

*** CALCULATE THE TOTAL PRESSURE LOSS IN THE TUBES

PLTTOT = PLTF + PLTR

*** CALCULATE THE SHELL-SIDE FLUID FRICTION FACTOR WHICH
 *** DEPENDS ON THE REYNOLDS NUMBER TAKEN FROM A CORRELATION
 *** OF FRICTION FACTORS FOR TUBE BUNDLES WITH 25% CUT
 *** SEGMENTAL BAFFLES FROM KERN PAGE 838

CALL SFRICT(RES,SRE,SF,FRS)

*** CALCULATE THE SHELL-SIDE PRESSURE LOSS VIA EQUATION
 *** FROM KERN ON PAGE 147

PLSTOT = (FRS * L * (GS**2.0) *
 \$ (INT((L*12.0) / (DSI(N) * PICKS)) + 1.0) * 0.006944)
 \$ / (2.0 * AG * DENS * (DEQS/12.0) * ((DVS/DVSF) ** 0.14))

*** PRINT THE FINAL CONFIGURATION, LENGTH , AND PRESSURE
 *** DROP FOR THE HEAT EXCHANGER

PRINT*,
 PRINT*, 'THE FINAL CONFIGURATION IS :'
 PRINT*,
 WRITE(*,601) DSI(N)
 601 FORMAT ('SHELL DIAMETER: ', F6.1, ' INCHES')
 PRINT*,
 WRITE(*,602) NP
 602 FORMAT ('NUMBER OF TUBE-SIDE PASSES: ', F3.1)
 PRINT*,
 WRITE(*,603) PICKS
 603 FORMAT ('BAFFLE SPACING: ', F5.2, ' OR')
 PRINT*,

*** USING THE CALCULATED VALUE OF THE OVERALL HEAT TRANSFER
*** COEFFICIENT, CALCULATE THE REQUIRED HEAT TRANSFER AREA.

$$ACORR = (NTU * CMIN) / U_{CALC}$$

*** USING THE CORRECT VALUE OF REQUIRED HEAT TRANSFER AREA,
*** CALCULATE THE LENGTH OF TUBING OF THE DESIRED HEAT
*** EXCHANGER CONFIGURATION.

IF (PICKP .EQ. 2.0) THEN
L = ACORR / AL2(N)
END IF

IF (PICKP .EQ. 4.0) THEN
L = ACORR / AL4(N)
END IF

IF (PICKP .EQ. 6.0) THEN
L = ACORR / AL6(N)
END IF

IF (PICKP .EQ. 8.0) THEN
L = ACORR / AL8(N)
END IF

NP = PICKP
AG = 32.1740

*** CALCULATE THE ESTIMATED COST OF THE HEAT EXCHANGER DESIGN
*** USING EQUATIONS FOUND IN THE BOOK - EFFICIENCY AND COSTING
*** BY GAGGIOLI ON PAGE 232

IF (PICKP .EQ. 2.0) THEN
Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
\$ ((TPH/12.0) ** 0.75) * (NT2(N) ** 0.975)
END IF

IF (PICKP .EQ. 4.0) THEN
Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
\$ ((TPH/12.0) ** 0.75) * (NT4(N) ** 0.975)
END IF

IF (PICKP .EQ. 6.0) THEN
Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
\$ ((TPH/12.0) ** 0.75) * (NT6(N) ** 0.975)
END IF

IF (PICKP .EQ. 8.0) THEN
Z = CF * 1150.0 * ((DSI(N)/12.0) ** 1.05) * (L ** 0.3) *
\$ ((TPH/12.0) ** 0.75) * (NT8(N) ** 0.975)
END IF

*** CALCULATE THE TUBE-SIDE FLUID FRICTION FACTOR WHICH

```

      TAVE = (TTAVE + TSAVE) / 2.0
      CALL VGAS (TAVE,DV)
      DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 4.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VDOWA (TAVE,DV)
  DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 5.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VKERO (TAVE,DV)
  DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 6.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VSAE (TAVE,DV)
  DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 7.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VEGLY (TAVE,DV)
  DVTF = DV
END IF

```

```

HI = (KT/(DTIN/12.0)) * 0.027 * (RET**0.8) *
$ ((CPT * DVT * 3600.0/KT)**0.3333) * ((DVT/DVTF)**0.14)

```

```

*** CALCULATE THE TUBE WALL RESISTANCE FROM EQUATION FOUND IN
*** TEMA STANDARDS HANDBOOK P 137.

```

```

RW = (DTOT/(24.0*KW)) * LOG (DTOT/(DTOT - 2.0*(DTOT-DTIN)))

```

```

*** CALCULATE THE AO/AI RATIO WHICH REDUCES TO DO/DI

```

```

TRATIO = DTOT / DTIN

```

```

*** CALCULATE THE OVERALL HEAT TRANSFER COEFFICIENT

```

```

IF (AFOUL .EQ. 'Y') THEN

```

```

      UCALC = 1.0/((1.0/HO) + RO + RW + (RI*TRATIO)
$      + ((1.0/HI)*TRATIO))

```

```

ELSE IF (AFOUL .EQ. 'N') THEN

```

```

      UCALC = 1.0/((1.0/HO) + RW + ((1.0/HI)*TRATIO))

```

```

END IF

```



```
TAVE = (TTAVE + TSAVE) / 2.0
CALL VMETH (TAVE,DV)
DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 3.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VGAS (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 4.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VDOWA (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 5.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VKERO (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 6.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VSAE (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 7.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VEGLY (TAVE,DV)
  DVSF = DV
END IF
```

```
HO = (KS/(DEQS/12.0)) * 0.36 * (RES**0.55) *
$ ((CPS * DVS + 3600.0/KS)**0.3333) * ((DVS/DVSF)**0.14)
```

```
*** FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
*** CORPORATION P 136
```

```
IF(PICKFS .EQ. 1.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VWATER (TAVE,DV)
  DVTF = DV
END IF
```

```
IF(PICKFS .EQ. 2.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VMETH (TAVE,DV)
  DVTF = DV
END IF
```

```
IF(PICKFS .EQ. 3.0) THEN
```

```

IF (PICKP .EQ. 8.0) THEN
  VT = (C*MT) / (ATIN * (NT8(N)/8.0) )
END IF

```

```

***  CALCULATE THE EQUIVALENT DIAMETER OF THE FLUID CONDUIT
***  CROSSING THE TUBES USING EQUATIONS FROM KERN IN HIS BOOK
***  PROCESS HEAT TRANSFER ON PAGE 139. THE EQUATION IS FOR A
***  TRIANGULAR PITCH ARRANGEMENT ONLY. UNITS ARE IN INCHES.

```

```

DEQS=(4.0*((.5*TPH)*(.86*TPH)-((.5*PI*(DTOT**2.))/4.0)))
$  /( (.5*DTOT) )

```

```

***  THE MASS VELOCITY OF THE SHELL FLUID IS CALCULATED. UNITS
***  ARE LB/SEC-SQUARE FEET

```

```

GS = VS * DENS

```

```

***  THE REYNOLDS NUMBER OF THE SHELL FLUID IS CALCULATED

```

```

RES = ( (DEQS / 12.0) * GS ) / DVS

```

```

***  THE MASS VELOCITY OF THE TUBE FLUID IS CALCULATED. UNITS
***  ARE LB/SEC-SQUARE FEET

```

```

GT = VT * DENT

```

```

***  THE REYNOLDS NUMBER OF THE TUBE FLUID IS CALCULATED

```

```

RET = ( (DTIN / 12.0) * GT ) / DVT

```

```

***  NOW THAT THE CONFIGURATION HAS BEEN CHOSEN, CALCULATE THE
***  OVERALL U VALUE FOR THE HEAT EXCHANGER USING GENERALIZED
***  CORRELATIONS FOUND IN THE REFERENCES.

```

```

***  FROM TEMA PAGE 137, THE OVERALL HEAT TRANSFER COEFFICIENT
***  REQUIRES THE EVALUATION OF:
***  HO : THE FILM COEFFICIENT OF THE FLUID OUTSIDE THE TUBES
***  HI : THE FILM COEFFICIENT OF THE FLUID INSIDE THE TUBES
***  RO : THE FOULING RESISTANCE ON THE OUTSIDE OF THE TUBES
***  RI : THE FOULING RESISTANCE ON THE INSIDE OF THE TUBES
***  RW : THE RESISTANCE OF THE TUBE WALL REFERRED TO THE
***  OUTSIDE SURFACE OF THE TUBE WALL
***  AO/AI : THE RATIO OF OUTSIDE TO INSIDE SURFACE OF TUBING

```

```

***  FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
***  CORPORATION P 157

```

```

IF(PICKFS .EQ. 1.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VWATER (TAVE,DV)
  DVSF = DV
END IF

```

```

IF(PICKFS .EQ. 2.0) THEN

```

*** IF THERE ARE AVAILABLE CONFIGURATIONS THAT MEET THE
 *** DESIGN SPECIFICATIONS, THE USER IS ASKED FOR HIS
 *** CHOICE OF CONFIGURATIONS THAT A FINAL OUTPUT IS DESIRED

```

70 PRINT*, ' '
   PRINT*, ' '
   PRINT*, 'THE NEXT THREE QUESTIONS PERTAIN TO YOUR CHOICE'
   PRINT*, 'OF THE ABOVE LISTED HEAT EXCHANGER '
   PRINT*, 'CONFIGURATIONS. REMEMBER ONLY THESE LISTED'
   PRINT*, 'CONFIGURATIONS WILL MEET THE REQUIREMENTS YOU'
   PRINT*, 'HAVE INPUT INTO THE PROGRAM'
   PRINT*, ' '
71 PRINT*, 'WHAT SHELL DIAMETER DO YOU CHOOSE ? TYPE 12.0,'
   PRINT*, '14.0, 28.0 etc.'
   PRINT*, ' '
   READ*, PICKD
   PRINT*, ' '
   PRINT*, 'WHAT NUMBER OF TUBE PASSES DO YOU CHOOSE ? TYPE'
   PRINT*, '2.0, 4.0, 6.0, OR 8.0'
   PRINT*, ' '
   READ*, PICKP
   PRINT*, ' '
   PRINT*, 'WHAT SHELL BAFFLE SPACING DO YOU CHOOSE ? TYPE'
   PRINT*, '0.2, 0.45, OR 1.0'
   PRINT*, ' '
   READ*, PICKS

```

$N = \text{INT}((\text{PICKD}/2.0) - 2.0)$

*** ONCE THE HEAT EXCHANGER CONFIGURATION HAS BEEN SELECTED, THE
 *** PROGRAM CAN NOW CALCULATE THE ACTUAL SHELL AND TUBE FLUID
 *** VELOCITIES. NOTE THAT C IS A CONVERSION FACTOR TO GET VELOCITY
 *** IN FT/SEC BY INPUTTING A FLOW RATE OF GALLONS PER MINUTE AND
 *** HAVING AREA IN INCHES.

$C = .3208342$

$ASC = (DSIU(N)/TPH) * TC * (PICKS * DSI(N))$

$VS = C * MS/ASC$

```

IF (PICKP .EQ. 2.0) THEN
  VT = (C*MT) / (ATIN * (NT2(N)/2.0) )
END IF

```

```

IF (PICKP .EQ. 4.0) THEN
  VT = (C*MT) / (ATIN * (NT4(N)/4.0) )
END IF

```

```

IF (PICKP .EQ. 6.0) THEN
  VT = (C*MT) / (ATIN * (NT6(N)/6.0) )
END IF

```

```

PICKP = 8.0
N = I
CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

IF (OKFIN .EQ. 1) THEN
    CALL PRINT8 (J,J8,PICKS,PICKD)
END IF

ELSE IF (M3 .EQ. 1) THEN

    PICKD = (I + 2.0) * 2.0
    PICKS = 1.0
    PICKP = 8.0
    N = I
    CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

    IF (OKFIN .EQ. 1) THEN
        CALL PRINT8 (J,J8,PICKS,PICKD)
    END IF

END IF

66 CONTINUE

IF (J8 .EQ. 0) THEN
    PRINT*, ' '
    PRINT*, 'NO 8-PASS HEAT EXCHANGER CONFIGURATIONS'
    PRINT*, 'WILL MEET THE PROBLEM SPECIFICATIONS.'
END IF

*** IF NO SHELL CONFIGURATIONS MEET THE CRITERIA THE PROGRAM
*** MOVES TO THE END AND ASKS IF ANY MORE DESIGNS ARE TO
*** BE COMPUTED

IF (J2 .EQ. 0 .AND. J4 .EQ. 0 .AND. J6 .EQ. 0 .AND.
$ J8 .EQ. 0) THEN
    PRINT*, ' '
    PRINT*, 'NO AVAILABLE SHELL DIAMETERS WILL MEET THE '
    PRINT*, 'LENGTH AND PRESSURE RESTRICTIONS GIVEN IN THIS'
    PRINT*, 'PROBLEM. SPLIT SHELL DESIGN OR MULTIPLE SHELLS'
    PRINT*, 'IN SERIES MUST BE USED. THIS COMPUTER PROGRAM'
    PRINT*, 'IS NOT SET UP TO HANDLE THAT TYPE OF HEAT '
    PRINT*, 'EXCHANGER DESIGN.'
    PRINT*, ' '
    GO TO 700
END IF

```

M1 = 0
M2 = 0
M3 = 0

*** FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
*** RECOMMENDED MINIMUMS

IF(MT .GE. QTL8(I) .AND. MT .LE. QTH8(I)) THEN
K8 = 1
END IF

IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
M1 = 1
END IF

IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
M2 = 1
END IF

IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
M3 = 1
END IF

IF(K8 .EQ. 0) THEN
GO TO 66
END IF

IF(M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
GO TO 66
END IF

*** A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
*** THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

IF (M1 .EQ. 1) THEN

PICKD = (I + 2.0) * 2.0
PICKS = 0.2
PICKP = 8.0
N = I

CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
\$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
\$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
\$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
\$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

IF (OKFIN .EQ. 1) THEN
CALL PRINT8 (J,J8,PICKS,PICKD)
END IF

ELSE IF (M2 .EQ. 1) THEN

PICKD = (I + 2.0) * 2.0
PICKS = 0.45

```

      IF (OKFIN .EQ. 1) THEN
        CALL PRINT6 (J,J6,PICKS,PICKD)
      END IF

```

```

    ELSE IF (M2 .EQ. 1) THEN

```

```

      PICKD = (I + 2.0) * 2.0
      PICKS = 0.45
      PICKP = 6.0
      N = I

```

```

      CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$      PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$      DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$      AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$      SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

```

```

      IF (OKFIN .EQ. 1) THEN
        CALL PRINT6 (J,J6,PICKS,PICKD)
      END IF

```

```

    ELSE IF (M3 .EQ. 1) THEN

```

```

      PICKD = (I + 2.0) * 2.0
      PICKS = 1.0
      PICKP = 6.0
      N = I

```

```

      CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$      PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$      DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$      AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$      SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

```

```

      IF (OKFIN .EQ. 1) THEN
        CALL PRINT6 (J,J6,PICKS,PICKD)
      END IF

```

```

    END IF

```

```

64  CONTINUE

```

```

      IF (J6 .EQ. 0) THEN
        PRINT*, ' '
        PRINT*, 'NO 6-PASS HEAT EXCHANGER CONFIGURATIONS'
        PRINT*, 'WILL MEET THE PROBLEM SPECIFICATIONS.'
      END IF

```

```

      PRINT*, ' '

```

```

***  ALL AVAILABLE 8-PASS CONFIGURATIONS ARE CHECKED

```

```

65  DO 66, I = 1,28

```

```

      K3 = 0

```

```

END IF

PRINT*, ' '

*** ALL AVAILABLE 6-PASS CONFIGURATIONS ARE CHECKED

63 DO 64, I = 1,28

    K6 = 0
    M1 = 0
    M2 = 0
    M3 = 0

*** FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
*** RECOMMENDED MINIMUMS

    IF (MT .GE. QTL6(I) .AND. MT .LE. QTH6(I)) THEN
        K6 = 1
    END IF

    IF (MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
        M1 = 1
    END IF

    IF (MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
        M2 = 1
    END IF

    IF (MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
        M3 = 1
    END IF

    IF (K6 .EQ. 0) THEN
        GO TO 64
    END IF

    IF (M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
        GO TO 64
    END IF

*** A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
*** THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

    IF (M1 .EQ. 1) THEN

        PICKD = (I + 2.0) * 2.0
        PICKS = 0.2
        PICKP = 6.0
        N = 1
        CALL PRESCK (DSIU,TFH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$         PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$         DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$         AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$         SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

```

PICKD = (I + 2.0) * 2.0

PICKS = 0.2

PICKP = 4.0

N = I

CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
\$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
\$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
\$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
\$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

IF (OKFIN .EQ. 1) THEN

CALL PRINT4 (J,J4,PICKS,PICKD)

END IF

ELSE IF (M2 .EQ. 1) THEN

PICKD = (I + 2.0) * 2.0

PICKS = 0.45

PICKP = 4.0

N = I

CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
\$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
\$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
\$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
\$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

IF (OKFIN .EQ. 1) THEN

CALL PRINT4 (J,J4,PICKS,PICKD)

END IF

ELSE IF (M3 .EQ. 1) THEN

PICKD = (I + 2.0) * 2.0

PICKS = 1.0

PICKP = 4.0

N = I

CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
\$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
\$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
\$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
\$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

IF (OKFIN .EQ. 1) THEN

CALL PRINT4 (J,J4,PICKS,PICKD)

END IF

END IF

62 CONTINUE

IF (J4 .EQ. 0) THEN

PRINT*, ' '

PRINT*, 'NO 4-PASS HEAT EXCHANGER CONFIGURATIONS

PRINT*, 'WILL MEET THE PROBLEM SPECIFICATIONS.'


```

        END IF
60    CONTINUE

        IF (J2 .EQ. 0) THEN
            PRINT*, ' '
            PRINT*, 'NO 2-PASS HEAT EXCHANGER CONFIGURATIONS'
            PRINT*, 'WILL MEET THE PROBLEM SPECIFICATIONS.'
        END IF

        PRINT*, ' '

***    ALL AVAILABLE 4-PASS CONFIGURATIONS ARE CHECKED
61    DO 62, I = 1,28

        K4 = 0
        M1 = 0
        M2 = 0
        M3 = 0

***    FLOW RATES ARE CHECKED TO SEE IF THEY FALL WITHIN
***    RECOMMENDED MINIMUMS

        IF(MT .GE. QTL4(I) .AND. MT .LE. QTH4(I)) THEN
            K4 = 1
        END IF

        IF(MS .GE. QSK1L(I) .AND. MS .LE. QSK1H(I)) THEN
            M1 = 1
        END IF

        IF(MS .GE. QSK2L(I) .AND. MS .LE. QSK2H(I)) THEN
            M2 = 1
        END IF

        IF(MS .GE. QSK3L(I) .AND. MS .LE. QSK3H(I)) THEN
            M3 = 1
        END IF

        IF(K4 .EQ. 0) THEN
            GO TO 62
        END IF

        IF(M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
            GO TO 62
        END IF

***    A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
***    THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA

        IF (M1 .EQ. 1) THEN

```

```
IF (M1 .EQ. 0 .AND. M2 .EQ. 0 .AND. M3 .EQ. 0) THEN
  GO TO 60
END IF
```

```
*** A PRESSURE CHECK AND LENGTH CHECK SUBROUTINE IS CALLED IF
*** THE SHELL DIAMETER MEETS THE INITIAL FLOW RATE CRITERIA
```

```
IF (M1 .EQ. 1) THEN
```

```
PICKD = (I + 2.0) * 2.0
PICKS = 0.2
PICKP = 2.0
N = I
```

```
CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)
```

```
IF (OKFIN .EQ. 1) THEN
  CALL PRINT2 (J,J2,PICKS,PICKD)
END IF
```

```
ELSE IF (M2 .EQ. 1) THEN
```

```
PICKD = (I + 2.0) * 2.0
PICKS = 0.45
PICKP = 2.0
N = I
```

```
CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)
```

```
IF (OKFIN .EQ. 1) THEN
  CALL PRINT2 (J,J2,PICKS,PICKD)
END IF
```

```
ELSE IF (M3 .EQ. 1) THEN
```

```
PICKD = (I + 2.0) * 2.0
PICKS = 1.0
PICKP = 2.0
N = I
```

```
CALL PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$ PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$ DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$ AFOUL,RO,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$ SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)
```

```
IF (OKFIN .EQ. 1) THEN
  CALL PRINT2 (J,J2,PICKS,PICKD)
END IF
```

```

        GO TO 4
    END IF

    PRINT*, ' '
702  PRINT*, 'DO YOU WANT TO CHANGE THE TUBING MATERIAL ?'
    PRINT*, ' '

    READ (5,101) AMATE

        IF (AMATE .EQ. 'Y') THEN
            GO TO 703
        END IF

    PRINT*, ' '
704  PRINT*, 'DO YOU WANT TO CHANGE THE TUBE FLUID CHOICE ?'
    PRINT*, ' '

    READ (5,101) ATFLU

        IF (ATFLU .EQ. 'Y') THEN
            GO TO 705
        END IF

    PRINT*, ' '
706  PRINT*, 'DO YOU WANT TO CHANGE THE SHELL FLUID CHOICE ?'
    PRINT*, ' '

    READ (5,101) ASFLU

        IF (ASFLU .EQ. 'Y') THEN
            GO TO 707
        END IF

    PRINT*, ' '
708  PRINT*, 'DO YOU WANT TO CHANGE THE TUBE FLUID TEMPS ?'
    PRINT*, ' '

    READ (5,101) ATTEM

        IF (ATTEM .EQ. 'Y') THEN
            GO TO 709
        END IF

    PRINT*, ' '
710  PRINT*, 'DO YOU WANT TO CHANGE THE TUBE FLUID FLOW RATE ?'
    PRINT*, ' '

    READ (5,101) ATGPM

        IF (ATGPM .EQ. 'Y') THEN
            GO TO 711
        END IF

    PRINT*, ' '

```

```

712 PRINT*, 'DO YOU WANT TO CHANGE THE SHELL FLUID TEMP ?'
    PRINT*, ' '

    READ (5,101) ASTEM

    IF (ASTEM .EQ. 'Y') THEN
        GO TO 713
    END IF

    PRINT*, ' '
714 PRINT*, 'DO YOU WANT TO CHANGE THE SHELL FLUID FLOW RATE ?'
    PRINT*, ' '

    READ (5,101) ASGPM

    IF (ASGPM .EQ. 'Y') THEN
        GO TO 715
    END IF

    PRINT*, ' '
716 PRINT*, 'DO YOU WANT TO CHANGE THE DECISION ON WHETHER OR'
    PRINT*, 'NOT A FOULING FACTOR IS USED ?'
    PRINT*, ' '

    READ (5,101) AFFAC

    PRINT*, ' '
    PRINT*, 'DO YOU WANT TO CHANGE THE LENGTH RESTRICTION ?'
    PRINT*, ' '

    READ (5,101) ALREST

    PRINT*, ' '
    PRINT*, 'DO YOU WANT TO CHANGE THE MAXIMUM ALLOWABLE'
    PRINT*, 'PRESSURE LOSSES ?'
    PRINT*, ' '

    READ (5,101) APRES

    GO TO 3

    END IF

    PRINT*, ' '
800 PRINT*, 'DO YOU WANT TO DO ANY OTHER HEAT EXCHANGER '
    PRINT*, 'PROBLEMS ? TYPE Y FOR YES AND N FOR NO.'

    READ (5,101) ACALC

    IF (ACALC .EQ. 'N') THEN
        GO TO 999
    END IF

    IF (ACALC .EQ. 'Y') THEN

```

AGAIN = 'N'
AFFAC = 'Y'
ALREST = 'Y'
APRES = 'Y'
GO TO 4

END IF

999 STOP 'DONE'

END

APPENDIX C

FLUID PROPERTY SUBROUTINES (WATER ONLY)

SUBROUTINE DWATER(TAVE,DEN)
*** THIS SUBROUTINE CALCULATES THE DENSITY OF WATER AT
*** THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
*** FROM 32.0 DEG F TO 600 DEG F. UNITS ARE LBM/CUBIC FT

IF (TAVE .LE. 32.0) THEN
DEN = 62.4
GO TO 100
END IF

IF (TAVE .LE. 50.0) THEN
DEN = 62.4
GO TO 100
END IF

IF (TAVE .EQ. 100.0) THEN
DEN = 62.0
GO TO 100
END IF

IF (TAVE .LT. 100.0) THEN
T = 50.0
T1 = 100.0
D = 62.4
D1 = 62.0
FRAC = (100.0 - TAVE) / (T1 - T) * (D - D1)
DEN = D1 + FRAC
GO TO 100
END IF

IF (TAVE .EQ. 150.0) THEN
DEN = 61.2
GO TO 100
END IF

IF (TAVE .LT. 150.0) THEN
T = 100.0
T1 = 150.0
D = 62.0
D1 = 61.2
FRAC = (150.0 - TAVE) / (T1 - T) * (D - D1)
DEN = D1 + FRAC
GO TO 100
END IF

IF (TAVE .EQ. 200.0) THEN
DEN = 60.1
GO TO 100
END IF

IF (TAVE .LT. 200.0) THEN
T = 150.0
T1 = 200.0

```

      D = 61.2
      D1 = 60.1
      FRAC = (200.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
END IF

IF (TAVE .EQ. 250.0) THEN
      DEN = 58.8
      GO TO 100
END IF

IF (TAVE .LT. 250.0) THEN
      T = 200.0
      T1 = 250.0
      D = 60.1
      D1 = 58.8
      FRAC = (250.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
END IF

IF (TAVE .EQ. 300.0) THEN
      DEN = 57.3
      GO TO 100
END IF

IF (TAVE .LT. 300.0) THEN
      T = 250.0
      T1 = 300.0
      D = 58.8
      D1 = 57.3
      FRAC = (300.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
END IF

IF (TAVE .EQ. 350.0) THEN
      DEN = 55.6
      GO TO 100
END IF

IF (TAVE .LT. 350.0) THEN
      T = 300.0
      T1 = 350.0
      D = 57.3
      D1 = 55.6
      FRAC = (350.0 - TAVE) / (T1 - T) * (D - D1)
      DEN = D1 + FRAC
      GO TO 100
END IF

IF (TAVE .EQ. 400.0) THEN
      DEN = 53.6

```


GO TO 100
END IF

IF (TAVE .LT. 400.0) THEN
T = 350.0
T1 = 400.0
D = 55.6
D1 = 53.6
FRAC = (400.0 - TAVE) / (T1 - T) * (D - D1)
DEN = D1 + FRAC
GO TO 100
END IF

IF (TAVE .EQ. 450.0) THEN
DEN = 51.6
GO TO 100
END IF

IF (TAVE .LT. 450.0) THEN
T = 400.0
T1 = 450.0
D = 53.6
D1 = 51.6
FRAC = (450.0 - TAVE) / (T1 - T) * (D - D1)
DEN = D1 + FRAC
GO TO 100
END IF

IF (TAVE .EQ. 500.0) THEN
DEN = 49.0
GO TO 100
END IF

IF (TAVE .LT. 500.0) THEN
T = 450.0
T1 = 500.0
D = 51.6
D1 = 49.0
FRAC = (500.0 - TAVE) / (T1 - T) * (D - D1)
DEN = D1 + FRAC
GO TO 100
END IF

IF (TAVE .EQ. 550.0) THEN
DEN = 45.9
GO TO 100
END IF

IF (TAVE .LT. 550.0) THEN
T = 500.0
T1 = 550.0
D = 49.0
D1 = 45.9
FRAC = (550.0 - TAVE) / (T1 - T) * (D - D1)

```
DEN = D1 + FRAC  
GO TO 100  
END IF
```

```
IF (TAVE .EQ. 600.0) THEN  
DEN = 42.4  
GO TO 100  
END IF
```

```
IF (TAVE .LT. 600.0) THEN  
T = 550.0  
T1 = 600.0  
D = 45.9  
D1 = 42.4  
FRAC = (600.0 - TAVE) / (T1 - T) * (D - D1)  
DEN = D1 + FRAC  
GO TO 100  
END IF
```

```
IF (TAVE .GT. 600.0) THEN  
DEN = 42.4  
END IF
```

```
100 RETURN  
END
```

SUBROUTINE CPWATR(TAVE,CP)
*** THIS SUBROUTINE CALCULATES THE SPECIFIC HEAT OF WATER
*** AT THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
*** FROM 32.0 DEG F TO 600 DEG F. UNITS ARE BTU/LBM-DEG F

IF (TAVE .LE. 32.0) THEN
 CP = 1.01
 GO TO 101
END IF

IF (TAVE .EQ. 50.0) THEN
 CP = 1.00
 GO TO 101
END IF

IF (TAVE .LT. 50.0) THEN
 T = 32.0
 T1 = 50.0
 C = 1.01
 C1 = 1.00
 FRAC = (50.0 - TAVE) / (T1 - T) * (C - C1)
 CP = C1 + FRAC
 GO TO 101
END IF

IF (TAVE .EQ. 100.0) THEN
 CP = 0.998
 GO TO 101
END IF

IF (TAVE .LT. 100.0) THEN
 T = 50.0
 T1 = 100.0
 C = 1.00
 C1 = 0.998
 FRAC = (100.0 - TAVE) / (T1 - T) * (C - C1)
 CP = C1 + FRAC
 GO TO 101
END IF

IF (TAVE .EQ. 150.0) THEN
 CP = 1.00
 GO TO 101
END IF

IF (TAVE .LT. 150.0) THEN
 T = 100.0
 T1 = 150.0
 C = 0.998
 C1 = 1.00
 FRAC = (150.0 - TAVE) / (T1 - T) * (C1 - C)
 CP = C1 - FRAC
 GO TO 101
END IF

```
IF (TAVE .LE. 200.0) THEN  
  CP = 1.00  
  GO TO 101  
END IF
```

```
IF (TAVE .EQ. 250.0) THEN  
  CP = 1.01  
  GO TO 101  
END IF
```

```
IF (TAVE .LT. 250.0) THEN  
  T = 200.0  
  T1 = 250.0  
  C = 1.00  
  C1 = 1.01  
  FRAC = (250.0 - TAVE) / (T1 - T) * (C1 - C)  
  CP = C1 - FRAC  
  GO TO 101  
END IF
```

```
IF (TAVE .EQ. 300.0) THEN  
  CP = 1.03  
  GO TO 101  
END IF
```

```
IF (TAVE .LT. 300.0) THEN  
  T = 250.0  
  T1 = 300.0  
  C = 1.01  
  C1 = 1.03  
  FRAC = (300.0 - TAVE) / (T1 - T) * (C1 - C)  
  CP = C1 - FRAC  
  GO TO 101  
END IF
```

```
IF (TAVE .EQ. 350.0) THEN  
  CP = 1.05  
  GO TO 101  
END IF
```

```
IF (TAVE .LT. 350.0) THEN  
  T = 300.0  
  T1 = 350.0  
  C = 1.03  
  C1 = 1.05  
  FRAC = (350.0 - TAVE) / (T1 - T) * (C1 - C)  
  CP = C1 - FRAC  
  GO TO 101  
END IF
```

```
IF (TAVE .EQ. 400.0) THEN  
  CP = 1.08  
  GO TO 101
```

END IF

IF (TAVE .LT. 400.0) THEN

T = 350.0

T1 = 400.0

C = 1.05

C1 = 1.08

FRAC = (400.0 - TAVE) / (T1 - T) * (C1 - C)

CP = C1 - FRAC

GO TO 101

END IF

IF (TAVE .EQ. 450.0) THEN

CP = 1.12

GO TO 101

END IF

IF (TAVE .LT. 450.0) THEN

T = 400.0

T1 = 450.0

C = 1.08

C1 = 1.12

FRAC = (450.0 - TAVE) / (T1 - T) * (C1 - C)

CP = C1 - FRAC

GO TO 101

END IF

IF (TAVE .EQ. 500.0) THEN

CP = 1.19

GO TO 101

END IF

IF (TAVE .LT. 500.0) THEN

T = 450.0

T1 = 500.0

C = 1.12

C1 = 1.19

FRAC = (500.0 - TAVE) / (T1 - T) * (C1 - C)

CP = C1 - FRAC

GO TO 101

END IF

IF (TAVE .EQ. 550.0) THEN

CP = 1.31

GO TO 101

END IF

IF (TAVE .LT. 550.0) THEN

T = 500.0

T1 = 550.0

C = 1.19

C1 = 1.31

FRAC = (550.0 - TAVE) / (T1 - T) * (C1 - C)

CP = C1 - FRAC

GO TO 101
END IF

IF (TAVE .EQ. 600.0) THEN
CP = 1.51
GO TO 101
END IF

IF (TAVE .LT. 600.0) THEN
T = 550.0
T1 = 600.0
C = 1.31
C1 = 1.51
FRAC = (600.0 - TAVE) / (T1 - T) * (C1 - C)
CP = C1 - FRAC
GO TO 101
END IF

IF (TAVE .GT. 600.0) THEN
CP = 1.51
END IF

101 RETURN
END

SUBROUTINE KWATER(TAVE,K)
*** THIS SUBROUTINE CALCULATES THE THERMAL CONDUCTIVITY OF
*** WATER AT THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
*** FROM 32.0 DEG F TO 600 DEG F. UNITS ARE BTU/HR-FT-DEG F

REAL K, K1

IF (TAVE .LE. 32.0) THEN
K = 0.319
GO TO 107
END IF

IF (TAVE .EQ. 50.0) THEN
K = 0.332
GO TO 107
END IF

IF (TAVE .LT. 50.0) THEN
T = 32.0
T1 = 50.0
K = 0.319
K1 = 0.332
FRAC = (50.0 - TAVE) / (T1 - T) * (K1 - K)
K = K1 - FRAC
GO TO 107
END IF

IF (TAVE .EQ. 100.0) THEN
K = 0.364
GO TO 107
END IF

IF (TAVE .LT. 100.0) THEN
T = 50.0
T1 = 100.0
K = 0.332
K1 = 0.364
FRAC = (100.0 - TAVE) / (T1 - T) * (K1 - K)
K = K1 - FRAC
GO TO 107
END IF

IF (TAVE .EQ. 150.0) THEN
K = 0.384
GO TO 107
END IF

IF (TAVE .LT. 150.0) THEN
T = 100.0
T1 = 150.0
K = 0.364
K1 = 0.384
FRAC = (150.0 - TAVE) / (T1 - T) * (K1 - K)
K = K1 - FRAC

GO TO 107
END IF

IF (TAVE .EQ. 200.0) THEN
K = 0.394
GO TO 107
END IF

IF (TAVE .LT. 200.0) THEN
T = 150.0
T1 = 200.0
K = 0.384
K1 = 0.394
$$\text{FRAC} = (200.0 - \text{TAVE}) / (T1 - T) * (K1 - K)$$
$$K = K1 - \text{FRAC}$$

GO TO 107
END IF

IF (TAVE .EQ. 250.0) THEN
K = 0.396
GO TO 107
END IF

IF (TAVE .LT. 250.0) THEN
T = 200.0
T1 = 250.0
K = 0.394
K1 = 0.396
$$\text{FRAC} = (250.0 - \text{TAVE}) / (T1 - T) * (K1 - K)$$
$$K = K1 - \text{FRAC}$$

GO TO 107
END IF

IF (TAVE .EQ. 300.0) THEN
K = 0.395
GO TO 107
END IF

IF (TAVE .LT. 300.0) THEN
T = 250.0
T1 = 300.0
K = 0.396
K1 = 0.395
$$\text{FRAC} = (300.0 - \text{TAVE}) / (T1 - T) * (K - K1)$$
$$K = K1 + \text{FRAC}$$

GO TO 107
END IF

IF (TAVE .EQ. 350.0) THEN
K = 0.391
GO TO 107
END IF

IF (TAVE .LT. 350.0) THEN

APPENDIX E
TFRICT SUBROUTINE

PLTTOT = PLTF + PLTR

*** CALCULATE THE SHELL-SIDE FLUID FRICTION FACTOR WHICH
*** DEPENDS ON THE REYNOLDS NUMBER TAKEN FROM A CORRELATION
*** OF FRICTION FACTORS FOR TUBE BUNDLES WITH 25% CUT
*** SEGMENTAL BAFFLES FROM KERN PAGE 838

CALL SFRIC(T,RES,SRE,SF,FRS)

*** CALCULATE THE SHELL-SIDE PRESSURE LOSS VIA EQUATION
*** FROM KERN ON PAGE 147

PLSTOT = (FRS * L * (GS**2.0) *
\$ (INT((L*12.0) / (DSI(N) * PICKS)) + 1.0) * 0.006944)
\$ / (2.0 * AG * DENS *(DEQS/12.0)* ((DVS/DVSF) ** 0.14))

*** CHECK TO SEE IF THE PRESSURE LOSSES MEET THE RESTRICTIONS
*** IMPOSED BY THE USER

IF (PLTTOT .LT. AMAXTP) THEN
OKT = 1
ELSE
OKT = 0
END IF

IF (PLSTOT .LT. AMAXSP) THEN
OKS = 1
ELSE
OKS = 0
END IF

IF (OKL .EQ. 1 .AND. OKT .EQ. 1 .AND. OKS .EQ. 1) THEN
OKFIN = 1
ELSE
OKFIN = 0
END IF

RETURN
END

*** CALCULATE THE LENGTH OF TUBING OF THE DESIRED HEAT
*** EXCHANGER CONFIGURATION.

IF (PICKP .EQ. 2.0) THEN
L = ACORR / AL2(N)
END IF

IF (PICKP .EQ. 4.0) THEN
L = ACORR / AL4(N)
END IF

IF (PICKP .EQ. 6.0) THEN
L = ACORR / AL6(N)
END IF

IF (PICKP .EQ. 8.0) THEN
L = ACORR / AL8(N)
END IF

*** CHECK TO SEE IF THE ACTUAL DESIGN LENGTH IS LESS THAN THE
*** LENGTH RESTRICTION IMPOSED BY THE USER.

IF (L .LT. LRESTR) THEN
OKL = 1
ELSE
OKL = 0
END IF

NP = PICKP
AG = 32.1740

*** CALCULATE THE TUBE-SIDE FLUID FRICTION FACTOR WHICH
*** DEPENDS ON THE REYNOLDS NUMBER USING SEIDER AND TATE
*** FRICTION FACTOR CORRELATIONS FROM PAGE 836 OF KERN

CALL TFRIC(TRE,TRE,TF,FRT)

*** CALCULATE THE PRESSURE LOSSES IN THE TUBE DUE TO BOTH
*** THE FRICTION LOSSES AND THE RETURN LOSSES USING
*** EQUATIONS FROM PAGE 148 AND 836 OF KERN. NOTE THAT
*** .006944 IS A CONVERSION FACTOR FOR PSF TO PSI AND
*** .4335 IS A CONVERSION FACTOR FOR FEET OF WATER TO PSI

TAVE = TTAVE
CALL DWATER(TAVE,DEN)
DENW = DEN

PLTF = ((FRT*NP*L* (GT**2.0)) / ((DTIN/12.0)* 2.0 * AG
\$ * DENT * ((DVT/DVTF)**0.14))) * 0.006944

PLTR = ((2.0 * NP * (VT**2.0)) / (2.0 * AG *
\$ (DENT/DENW))) * 0.4335

*** CALCULATE THE TOTAL PRESSURE LOSS IN THE TUBES

```

      TAVE = (TTAVE + TSAVE) / 2.0
      CALL VDOWA (TAVE,DV)
      DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 5.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VKERO (TAVE,DV)
  DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 6.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VSAE (TAVE,DV)
  DVTF = DV
END IF

```

```

IF(PICKFS .EQ. 7.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VEGLY (TAVE,DV)
  DVTF = DV
END IF

```

```

HI = (KT/(DTIN/12.0)) * 0.027 * (RET**0.8) *
$ ((CPT * DVT * 3600.0/KT)**0.3333) * ((DVT/DVTF)**0.14)

```

```

*** CALCULATE THE TUBE WALL RESISTANCE FROM EQUATION FOUND IN
*** TEMA STANDARDS HANDBOOK P 137.

```

```

RW = (DTOT/(24.0*KW)) * LOG (DTOT/(DTOT - 2.0*(DTOT-DTIN)))

```

```

*** CALCULATE THE AO/AI RATIO WHICH REDUCES TO DO/DI

```

```

TRATIO = DTOT / DTIN

```

```

*** CALCULATE THE OVERALL HEAT TRANSFER COEFFICIENT

```

```

IF (AFOUL .EQ. 'Y') THEN

```

```

      UCALC = 1.0/((1.0/HO) + RO + RW + (RI*TRATIO)
$      + ((1.0/HI)*TRATIO))

```

```

ELSE IF (AFOUL .EQ. 'N') THEN

```

```

      UCALC = 1.0/((1.0/HO) + RW + ((1.0/HI)*TRATIO))

```

```

END IF

```

```

*** USING THE CALCULATED VALUE OF THE OVERALL HEAT TRANSFER
*** COEFFICIENT, CALCULATE THE REQUIRED HEAT TRANSFER AREA.

```

```

ACORR = (NTU * CMIN) / UCALC

```

```

*** USING THE CORRECT VALUE OF REQUIRED HEAT TRANSFER AREA,

```

```
TAVE = (TTAVE + TSAVE) / 2.0
CALL VGAS (TAVE,DV)
DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 4.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VDOWA (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 5.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VKERO (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 6.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VSAE (TAVE,DV)
  DVSF = DV
END IF
```

```
IF(PICKFS .EQ. 7.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VEGLY (TAVE,DV)
  DVSF = DV
END IF
```

```
HO = (KS/(DEQS/12.0)) * 0.36 * (RES**0.55) *
$ ((CPS * DVS * 3600.0/KS)**0.3333) * ((DVS/DVSF)**0.14)
```

```
*** FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
*** CORPORATION P 136
```

```
IF(PICKFS .EQ. 1.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VWATER (TAVE,DV)
  DVTF = DV
END IF
```

```
IF(PICKFS .EQ. 2.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VMETH (TAVE,DV)
  DVTF = DV
END IF
```

```
IF(PICKFS .EQ. 3.0) THEN
  TAVE = (TTAVE + TSAVE) / 2.0
  CALL VGAS (TAVE,DV)
  DVTF = DV
END IF
```

```
IF(PICKFS .EQ. 4.0) THEN
```

*** PROCESS HEAT TRANSFER ON PAGE 139. THE EQUATION IS FOR A
*** TRIANGULAR PITCH ARRANGEMENT ONLY. UNITS ARE IN INCHES.

$$DEQS = (4.0 * ((.5 * TPH) * (.86 * TPH) - ((.5 * PI * (DTOT ** 2.)) / 4.0))) / (.5 * DTOT)$$

*** THE VELOCITY OF THE SHELL FLUID IS CALCULATED. UNITS
*** ARE LB/SEC-SQUARE FEET

$$GS = VS * DENS$$

*** THE REYNOLDS NUMBER OF THE SHELL FLUID IS CALCULATED

$$RES = ((DEQS / 12.0) * GS) / DVS$$

*** THE MASS VELOCITY OF THE TUBE FLUID IS CALCULATED. UNITS
*** ARE LB/SEC-SQUARE FEET

$$GT = VT * DENT$$

*** THE REYNOLDS NUMBER OF THE TUBE FLUID IS CALCULATED

$$RET = ((DTIN / 12.0) * GT) / DVT$$

*** NOW THAT THE CONFIGURATION HAS BEEN CHOSEN, CALCULATE THE
*** OVERALL U VALUE FOR THE HEAT EXCHANGER USING GENERALIZED
*** CORRELATIONS FOUND IN THE REFERENCES.

*** FROM TEMA PAGE 137, THE OVERALL HEAT TRANSFER COEFFICIENT
*** REQUIRES THE EVALUATION OF:

*** HO : THE FILM COEFFICIENT OF THE FLUID OUTSIDE THE TUBES

*** HI : THE FILM COEFFICIENT OF THE FLUID INSIDE THE TUBES

*** RO : THE FOULING RESISTANCE ON THE OUTSIDE OF THE TUBES

*** RI : THE FOULING RESISTANCE ON THE INSIDE OF THE TUBES

*** RW : THE RESISTANCE OF THE TUBE WALL REFERRED TO THE

*** OUTSIDE SURFACE OF THE TUBE WALL

*** AO/AI : THE RATIO OF OUTSIDE TO INSIDE SURFACE OF TUBING

*** FROM 'HEAT EXCHANGE TECHNOLOGY' BY STRUTHERS WELLS
*** CORPORATION P 157

IF(PICKFS .EQ. 1.0) THEN

$$TAVE = (TTAVE + TSAVE) / 2.0$$

CALL VWATER (TAVE,DV)

$$DVSF = DV$$

END IF

IF(PICKFS .EQ. 2.0) THEN

$$TAVE = (TTAVE + TSAVE) / 2.0$$

CALL VMETH (TAVE,DV)

$$DVSF = DV$$

END IF

IF(PICKFS .EQ. 3.0) THEN

```

SUBROUTINE PRESCK (DSIU,TPH,TC,PICKS,DSI,MS,ASC,MT,ATIN,
$    PICKP,NT2,NT4,NT6,NT8,PI,DTOT,DENS,DVS,DENT,
$    DTIN,DVT,PICKFS,TTAVE,TSAVE,DV,KS,CPS,KT,CPT,KW,
$    AFOUL,R0,RW,RI,NTU,CMIN,AL2,AL4,AL6,AL8,TRE,TF,
$    SRE,SF,LRESTR,OKFIN,AMAXTP,AMAXSP,PICKD)

```

```

*** THIS SUBROUTINE DOES A PRESSURE CHECK ON BOTH THE TUBE
*** AND THE SHELL SIDES, AND ALSO DOES A LENGTH RESTRICTION
*** CHECK. IF THE SUBROUTINE RETURNS THE VALUE OF OKFIN = 1
*** THE CONFIGURATION MET THE CRITERIA. A VALUE OF OKFIN = 0
*** MEANS THAT THE CONFIGURATION FAILED ONE OR MORE OF THE
*** DESIGN CRITERIA IMPOSED BY THE USER.

```

```

DIMENSION DSIU(28),DSI(28),NT2(28),NT4(28),NT6(28),
$    NT8(28),AL2(28),AL4(28),AL6(28),AL8(28),TRE(34),
$    TF(34),SRE(34),SF(34)

```

```

CHARACTER*1 AFOUL
REAL NTU,MT,MS,LRESTR,NP,KS,KT,KW,L

```

```

N = INT( (PICKD/2.0) - 2.0 )

```

```

*** FOR EACH HEAT EXCHANGER CONFIGURATION , THE PROGRAM CAN NOW
*** THE ACTUAL SHELL AND TUBE FLUID VELOCITIES, AND DO A LENGTH
*** RESTRICTION CHECK, A SHELL PRESSURE LOSS CHECK, AND A TUBE
*** PRESSURE LOSS CHECK.
*** NOTE THAT C IS A CONVERSION FACTOR TO GET VELOCITY UNITS IN
*** FT/SEC BY INPUTTING A FLOW RATE OF GALLONS PER MINUTE AND
*** HAVING AREA IN INCHES.

```

```

C = .3208342

```

```

ASC = (DSIU(N)/TPH) * TC * (PICKS * DSI(N))

```

```

VS = C * MS/ASC

```

```

IF (PICKP .EQ. 2.0) THEN
    VT = (C*MT) / (ATIN * (NT2(N)/2.0) )
END IF

```

```

IF (PICKP .EQ. 4.0) THEN
    VT = (C*MT) / (ATIN * (NT4(N)/4.0) )
END IF

```

```

IF (PICKP .EQ. 6.0) THEN
    VT = (C*MT) / (ATIN * (NT6(N)/6.0) )
END IF

```

```

IF (PICKP .EQ. 8.0) THEN
    VT = (C*MT) / (ATIN * (NT8(N)/8.0) )
END IF

```

```

*** CALCULATE THE EQUIVALENT DIAMETER OF THE FLUID CONDUIT
*** CROSSING THE TUBES USING EQUATIONS FROM KERN IN HIS BOOK

```

APPENDIX D
PRESCK SUBROUTINE

DV = 0.000064
GO TO 102
END IF

IF (TAVE .LT. 550.0) THEN
T = 500.0
T1 = 550.0
V = 0.000071
V1 = 0.000064
FRAC = (550.0 - TAVE) / (T1 - T) * (V - V1)
DV = V1 + FRAC
GO TO 102
END IF

IF (TAVE .EQ. 600.0) THEN
DV = 0.000058
GO TO 102
END IF

IF (TAVE .LT. 600.0) THEN
T = 550.0
T1 = 600.0
V = 0.000064
V1 = 0.000058
FRAC = (600.0 - TAVE) / (T1 - T) * (V - V1)
DV = V1 + FRAC
GO TO 102
END IF

IF (TAVE .GT. 600.0) THEN
DV = 0.000058
END IF

102 RETURN
END

```

    T1 = 350.0
    V = 0.000126
    V1 = 0.000105
    FRAC = (350.0 - TAVE) / (T1 - T) * (V - V1)
    DV = V1 + FRAC
    GO TO 102
END IF

IF (TAVE .EQ. 400.0) THEN
    DV = 0.000091
    GO TO 102
END IF

IF (TAVE .LT. 400.0) THEN
    T = 350.0
    T1 = 400.0
    V = 0.000105
    V1 = 0.000091
    FRAC = (400.0 - TAVE) / (T1 - T) * (V - V1)
    DV = V1 + FRAC
    GO TO 102
END IF

IF (TAVE .EQ. 450.0) THEN
    DV = 0.000080
    GO TO 102
END IF

IF (TAVE .LT. 450.0) THEN
    T = 400.0
    T1 = 450.0
    V = 0.000091
    V1 = 0.000080
    FRAC = (450.0 - TAVE) / (T1 - T) * (V - V1)
    DV = V1 + FRAC
    GO TO 102
END IF

IF (TAVE .EQ. 500.0) THEN
    DV = 0.000071
    GO TO 102
END IF

IF (TAVE .LT. 500.0) THEN
    T = 450.0
    T1 = 500.0
    V = 0.000080
    V1 = 0.000071
    FRAC = (500.0 - TAVE) / (T1 - T) * (V - V1)
    DV = V1 + FRAC
    GO TO 102
END IF

IF (TAVE .EQ. 550.0) THEN

```

END IF

IF (TAVE .EQ. 200.0) THEN

DV = 0.000205

GO TO 102

END IF

IF (TAVE .LT. 200.0) THEN

T = 150.0

T1 = 200.0

V = 0.000292

V1 = 0.000205

FRAC = (200.0 - TAVE) / (T1 - T) * (V - V1)

DV = V1 + FRAC

GO TO 102

END IF

IF (TAVE .EQ. 250.0) THEN

DV = 0.000158

GO TO 102

END IF

IF (TAVE .LT. 250.0) THEN

T = 200.0

T1 = 250.0

V = 0.000205

V1 = 0.000158

FRAC = (250.0 - TAVE) / (T1 - T) * (V - V1)

DV = V1 + FRAC

GO TO 102

END IF

IF (TAVE .EQ. 300.0) THEN

DV = 0.000126

GO TO 102

END IF

IF (TAVE .LT. 300.0) THEN

T = 250.0

T1 = 300.0

V = 0.000158

V1 = 0.000126

FRAC = (300.0 - TAVE) / (T1 - T) * (V - V1)

DV = V1 + FRAC

GO TO 102

END IF

IF (TAVE .EQ. 350.0) THEN

DV = 0.000105

GO TO 102

END IF

IF (TAVE .LT. 350.0) THEN

T = 300.0

```
SUBROUTINE VWATER(TAVE,DV)
*** THIS SUBROUTINE CALCULATES THE DYNAMIC VISCOSITY OF
*** WATER AT THE AVERAGE FLUID TEMPERATURE FOR TEMPERATURES
*** FROM 32.0 DEG F TO 600 DEG F. UNITS ARE LB/FT-SEC
```

```
IF (TAVE .LE. 32.0) THEN
  DV = 0.0012
  GO TO 102
END IF
```

```
IF (TAVE .EQ. 50.0) THEN
  DV = 0.00088
  GO TO 102
END IF
```

```
IF (TAVE .LT. 50.0) THEN
  T = 32.0
  T1 = 50.0
  V = 0.0012
  V1 = 0.00088
  FRAC = (50.0 - TAVE) / (T1 - T) * (V - V1)
  DV = V1 + FRAC
  GO TO 102
END IF
```

```
IF (TAVE .EQ. 100.0) THEN
  DV = 0.000458
  GO TO 102
END IF
```

```
IF (TAVE .LT. 100.0) THEN
  T = 50.0
  T1 = 100.0
  V = 0.00088
  V1 = 0.000458
  FRAC = (100.0 - TAVE) / (T1 - T) * (V - V1)
  DV = V1 + FRAC
  GO TO 102
END IF
```

```
IF (TAVE .EQ. 150.0) THEN
  DV = 0.000292
  GO TO 102
END IF
```

```
IF (TAVE .LT. 150.0) THEN
  T = 100.0
  T1 = 150.0
  V = 0.000458
  V1 = 0.000292
  FRAC = (150.0 - TAVE) / (T1 - T) * (V - V1)
  DV = V1 + FRAC
  GO TO 102
```

IF (TAVE .EQ. 550.0) THEN

K = 0.325

GO TO 107

END IF

IF (TAVE .LT. 550.0) THEN

T = 500.0

T1 = 550.0

K = 0.349

K1 = 0.325

FRAC = (550.0 - TAVE) / (T1 - T) * (K - K1)

K = K1 + FRAC

GO TO 107

END IF

IF (TAVE .EQ. 600.0) THEN

K = 0.292

GO TO 107

END IF

IF (TAVE .LT. 600.0) THEN

T = 550.0

T1 = 600.0

K = 0.325

K1 = 0.292

FRAC = (600.0 - TAVE) / (T1 - T) * (K - K1)

K = K1 + FRAC

GO TO 107

END IF

IF (TAVE .GT. 600.0) THEN

K = 0.292

END IF

107 RETURN

END

```

      T = 300.0
      T1 = 350.0
      K = 0.395
      K1 = 0.391
      FRAC = (350.0 - TAVE) / (T1 - T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
    END IF

    IF (TAVE .EQ. 400.0) THEN
      K = 0.381
      GO TO 107
    END IF

    IF (TAVE .LT. 400.0) THEN
      T = 350.0
      T1 = 400.0
      K = 0.391
      K1 = 0.381
      FRAC = (400.0 - TAVE) / (T1 - T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
    END IF

    IF (TAVE .EQ. 450.0) THEN
      K = 0.367
      GO TO 107
    END IF

    IF (TAVE .LT. 450.0) THEN
      T = 400.0
      T1 = 450.0
      K = 0.381
      K1 = 0.367
      FRAC = (450.0 - TAVE) / (T1 - T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
    END IF

    IF (TAVE .EQ. 500.0) THEN
      K = 0.349
      GO TO 107
    END IF

    IF (TAVE .LT. 500.0) THEN
      T = 450.0
      T1 = 500.0
      K = 0.367
      K1 = 0.349
      FRAC = (500.0 - TAVE) / (T1 - T) * (K - K1)
      K = K1 + FRAC
      GO TO 107
    END IF

```

SUBROUTINE TFRICT(RET,TRE,TF,FRT)

*** THIS SUBROUTINE CALCULATES THE TUBE-SIDE FRICTION FACTOR
*** FOR USE IN THE DARCY-FANNING PRESSURE LOSS EQUATION FOR
*** THE TUBE-SIDE PRESSURE DROP DUE TO FRICTION. THE FACTOR
*** IS INTERPOLATED FROM VALUES TAKEN FROM A GRAPH OF TUBE-
*** SIDE FRICTION FACTORS PUBLISHED BY TEMA. THE FRICTION
*** FACTOR IS A DIMENSIONLESS PARAMETER.

DIMENSION TRE(34),TF(34)

I = 1
IF(RET .LE. TRE(I)) THEN
FRT = TF(I)
GO TO 104
END IF

DO 103, I= 2,34
IF(RET .LT. TRE(I)) THEN
FRAC = (TRE(I)-RET)/(TRE(I)-TRE(I-1))*(TF(I-1)-TF(I))
FRT = TF(I-1) - FRAC
GO TO 104
END IF

IF(RET .EQ. TRE(I)) THEN
FRT = TF(I)
GO TO 104
END IF

103 CONTINUE

PRINT*, ' THE TUBE-SIDE FLUID REYNOLDS NUMBER IS OUT OF '
PRINT*, ' RANGE OF THE PROGRAM. AN AVERAGE FRICTION '
PRINT*, ' FACTOR OF 0.035 WILL BE USED IN CALCULATIONS. '

FRT = 0.035

104 RETURN
END

APPENDIX F
SFRIC T SUBROUTINE

SUBROUTINE SFRIC(TRES,SRE,SF,FRS)

*** THIS SUBROUTINE CALCULATES THE SHELL-SIDE FRICTION FACTOR
*** FOR USE IN THE DARCY-FANNING PRESSURE LOSS EQUATION FOR
*** THE SHELL-SIDE PRESSURE DROP DUE TO FRICTION. THE FACTOR
*** IS INTERPOLATED FROM VALUES TAKEN FROM A GRAPH OF SHELL-
*** SIDE FRICTION FACTORS PUBLISHED BY KERN IN HIS BOOK PROCESS
*** HEAT TRANSFER. THE FRICTION FACTOR IS A DIMENSIONLESS
*** PARAMETER.

DIMENSION SRE(34),SF(34)

I = 1
IF(TRES .LE. SRE(I)) THEN
FRS = SF(I)
GO TO 106
END IF

DO 105, I= 2,34
IF(TRES .LT. SRE(I)) THEN
FRAC = (SRE(I)-TRES)/(SRE(I)-SRE(I-1))*(SF(I-1)-SF(I))
FRS = SF(I-1) - FRAC
GO TO 106
END IF

IF(TRES .EQ. SRE(I)) THEN
FRS = SF(I)
GO TO 106
END IF

105 CONTINUE

PRINT*, ' THE SHELL FLUID REYNOLDS NUMBER IS OUT OF'
PRINT*, 'RANGE OF THE PROGRAM. AN AVERAGE FRICTION'
PRINT*, 'FACTOR OF 0.30 WILL BE USED IN CALCULATIONS.'

FRS = 0.30

106 RETURN
END

APPENDIX G
PRINT SUBROUTINES

SUBROUTINE PRINT2(J,J2,PICKS,PICKD)

*** THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
*** CONFIGURATIONS FOR 2-PASS CONSTRUCTION

IF (J .EQ. 0) THEN

PRINT*, ' '

PRINT*, 'THE AVAILABLE HEAT EXCHANGER'

PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'

PRINT*, ' '

J = 1

END IF

IF (J2 .EQ. 0) THEN

PRINT*, ' '

PRINT*, 'THE AVAILABLE 2-PASS HEAT EXCHANGER'

PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'

PRINT*, ' '

J2 = 1

END IF

PRINT*, 'SHELL DIAMETER', PICKD, ' INCHES:'

PRINT*, 'WITH A ', PICKS, ' BAFFLE SPACING'

RETURN

END

SUBROUTINE PRINT4(J,J4,PICKS,PICKD)

*** THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
*** CONFIGURATIONS FOR 4-PASS CONSTRUCTION

IF (J .EQ. 0) THEN
 PRINT*, '
 PRINT*, 'THE AVAILABLE HEAT EXCHANGER'
 PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'
 PRINT*, '
 J = 1

END IF

IF (J4 .EQ. 0) THEN
 PRINT*, '
 PRINT*, 'THE AVAILABLE 4-PASS HEAT EXCHANGER'
 PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'
 PRINT*, '
 J4 = 1

END IF

PRINT*, 'SHELL DIAMETER', PICKD, ' INCHES:'
PRINT*, 'WITH A ', PICKS, ' BAFFLE SPACING'

RETURN
END

SUBROUTINE PRINT6(J,J6,PICKS,PICKD)

*** THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
*** CONFIGURATIONS FOR 6-PASS CONSTRUCTION

IF (J .EQ. 0) THEN
 PRINT*, ' '
 PRINT*, 'THE AVAILABLE HEAT EXCHANGER'
 PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'
 PRINT*, ' '
 J = 1

END IF

IF (J6 .EQ. 0) THEN
 PRINT*, ' '
 PRINT*, 'THE AVAILABLE 6-PASS HEAT EXCHANGER'
 PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'
 PRINT*, ' '
 J6 = 1

END IF

PRINT*, 'SHELL DIAMETER', PICKD, ' INCHES:'
PRINT*, 'WITH A ', PICKS, ' BAFFLE SPACING'

RETURN
END

SUBROUTINE PRINT8(J,J8,PICKS,PICKD)

*** THIS SUBPROGRAM PRINTS OUT THE AVAILABLE EXCHANGER
*** CONFIGURATIONS FOR 8-PASS CONSTRUCTION

IF (J .EQ. 0) THEN
 PRINT*, ' '
 PRINT*, 'THE AVAILABLE HEAT EXCHANGER'
 PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'
 PRINT*, ' '

J = 1

END IF

IF (J8 .EQ. 0) THEN
 PRINT*, ' '
 PRINT*, 'THE AVAILABLE 8-PASS HEAT EXCHANGER'
 PRINT*, 'CONFIGURATIONS ARE AS FOLLOWS:'
 PRINT*, ' '

J8 = 1

END IF

PRINT*, 'SHELL DIAMETER', PICKD, ' INCHES:'
PRINT*, 'WITH A ', PICKS, ' BAFFLE SPACING'

RETURN
END

END

FILMED

6-85

DTIC